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Wind-Powered Machines

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Redwood City, California

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16. Abstract In this book are presented the basic problems connected with the selection of layouts and calculation of parameters of wind machines, their energy-producing characteristics and technical and economic indices. Methods of optimal matching of wind engines with working machines, calculators for strength, and construction and automation of wind machines are analyzed in detail. A description is given of the setup of domestic and foreign wind installations for various purposes. Basic characteristics of wind as a source of energy, brief information from aerodynamics, the theory of the wind engine and calculation of its aerodynamic characteristics are presented. In conclusion, recommendations are presented for use of wind machines according to zone. The book is intended for engineers, designers and workers of scientific and research institutes connected with creation and utilization of wind machines and for engineers and mechanics in agriculture.

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Foreword

In recent years, in the USSR and in other countries, work on utilizing the energy of regenerative sources (wind, sun, geothermal waters, tides) has been broadened, primarily for regions not provided with a centralized power supply. Wind power has received the most widespread development.

Expanded scopes of electrification, including agriculture, have necessitated a new approach to the solution of a number of problems in wind energy. This was expressed in particular in the development of bases for optimal planning of machines which more fully corresponded to regional conditions, in the formulation of new principles and in creation of methods for calculating, designing and unifying wind engines, the utilization of more progressive types of drives and working machines combined with wind engines, and in automation of wind installations.

Principally new and more perfect machines for various purposes have been created in recent years on the basis of new investigations and inventions.

The optimal solution of many of the problems enumerated is made more complex by the fact that a wind machine works under complex conditions, and changing rates, depending on many factors. Design solutions and calculation of a number of elements of the machine are also relatively complex. The required attention was not paid to these peculiarities earlier, and a simplified approach to a number of problems connected with creation and utilization of wind machines occurred. The consequences of such an approach were frequently a reduction in economy of the machines and premature failure of them.

Wind machines have been most widely utilized for mechanization of the lifting of water in pastures, farms, reclaimed and sown areas, and also for charging storage batteries. This is the traditional direction for utilization of wind. During recent years, highly promising areas have appeared, where utilization of wind installations is also economically justified; these are, in particular, distillation of water, so-called cathode protection of pipelines and marine constructions from corrosion, vertical drainage of reclaimed land, and supplying power to independent special purpose installations (apparatuses for radio relay communications, automatic meteorological stations, lighthouses, beacons and others).

Although wind machines use a "free" source of energy, their effectiveness, determined by the costs presented, must not be lower than the effectiveness of other power installations, which can be used under actual conditions, they must also have a number of advantages over non-wind installations: longer service life, simpler servicing, etc.

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Wind installations must be wisely and organically inscribed in the overall system of energy supply with consideration for the specifics of the technical processes for which they are designed. In the first place, these are processes of agricultural production, where there are many widely spread objects requiring small amounts of power and located in areas to which access is difficult and which are removed from electrical power circuits. Therefore, installations with power up to 15 kW have received the most widespread development and will be the most promising in the coming period.

Concerning wind energy machines and stations (WES) of medium and high power, the author considers that the requirement for them is limited and that their usage is effective, as a rule, with grouped utilization of multicomponent systems or with the parallel operation of WES's with non-wind stations. These questions are an independent problem. Their solution requires development of new systems of automatic regulation, fundamentals of mechanical and electrical diagrams and creation of principally new designs for machines providing a significant reduction in metal consumption and cost and an essential increase in reliability.

The book reviews that which is essentially new, that which has been achieved in wind engineering and in the practice of utilizing wind machines over the past decade, and major attention is paid to the most promising low power machines. Specific questions connected with the creation of a WES are not illuminated in the book.

The circle of questions encompassing theories and practice of wind engineering and wind utilization is relatively wide. In this work, naturally, the task of examining all questions and problems connected with the utilization of the energy of wind is not assigned. The author considered it expedient to refrain from repeating a number of sections which are sufficiently fully expounded on in other books. In this work, he began from the understanding of the problem of wind energy noted above and strived to examine the problems having, in his opinion, the most important meaning for increasing effectiveness of wind machines and wind utilization as a whole. In accordance with this, more attention than is usual is devoted to calculation and design of machines and pumping windmills, and also to problems of energetics and engineering economics. Some information on the theory of the wind engine, aerodynamics and wind resources are presented briefly, in a scope necessary for understanding the basic content of the work. For readers interested by these questions in more detail, directions are given to sources presented in the bibliography.

The basis of the work consisted of investigations of the author and of the collective of the laboratory of the All-Union Scientific and Research Institute of Electrification of Agriculture (VIESKh) headed by him. The results of combined developments

with TsAGI [Central Institute of Aerohydrodynamics imeni N.Ye.Zhukovskiy] and other organizations were used, as were correlations of literary materials and data from government experiments. The results of a combined work of the author with I.V. Rozhdestvenskiy were used during the examination of some problems connected with the creation of electrical wind machines.

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WIND-POWERED MACHINES

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PART 1. RESOURCES OF WIND AND THEORY OF THE WIND ENGINE /6*

Chapter 1. The Wind as a Source of Energy

§ 1. Basic Characteristics of Wind

Wind -- the constant circulation of moving air masses in the atmosphere caused by unevenness in heating of the earth's surface by the sun -- is one of the most powerful sources of energy, which under favorable conditions can be used in the national economy. Wind-powered installations usually use the wind at heights of up to 70 m from the surface of the earth and therefore we will examine its characteristics in this stratum. At greater heights (8-12 km), in the so-called tropopause, more constant winds, called jet streams, take place. These winds are distinguished by high speed (25-80 m/sec), and their energy-producing value is significantly higher, but their characteristics differ essentially from the characteristics of surface wind. However, utilization of these streams is connected with surmounting serious engineering difficulties in the transfer of electrical energy to the earth from high altitudes and, probably, the problem of utilizing jet-stream energy cannot be practically solved earlier than over the next 7 to 10 years.

Wind is a complex geophysical process, and therefore its irregularity can be projected only with one degree of probability or another.

The most important energy-producing characteristic of wind, which evaluates its kinetic energy, is velocity. Under the influence of a number of meteorological factors (atmospheric disturbances, changes in solar activity and the quantity of thermal energy falling on the earth, etc.), as well as localized relief conditions, wind velocity changes in amount and direction. The direction of the velocity vector shows its angular position relative to the direction taken at the beginning of reading (usually, this is a northerly direction).

A Wilde vane or a low-inertia anemometer is used for measuring momentary wind velocity; the average value over an extended interval (several tenths of a second or minute) is measured with various

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* Numbers in the margin indicate pagination in the foreign text.

types of anemometers, which have indicator instruments and a registration part providing a record of velocities on tape. The velocity measuring error with an anemometer can comprise 5-7%, and therefore, if greater accuracy is required, for instance, during tests of wind engines and their models in aerodynamic tunnels, a /7 Pitot tube connected with a micromanometer is used [35].

While the momentary velocity of wind has an essential dynamic influence on a wind engine and affects the work of the automatic adjusting system, the development of energy depends first of all on the average velocity both with respect to time and to the area of the surface swept by the windmill.

The average wind velocity $\bar{v}$ over the time interval is determined by the relationship of the total measured values of momentary velocity v_i, to the number of measurements n over this time interval:

$$\bar{v} = \frac{\Sigma v_i}{n} \text{ m/sec} \quad (1)$$

Similarly, the average daily velocity $\bar{v}_{day}$ is defined as the quotient from division into 24 total average hourly velocities, and the average annual velocity is

$$\bar{v}_a = \frac{\Sigma v_{day}}{365} \text{ m/sec} \quad (2)$$

The average values for velocities in the region examined is determined according to observation data from meteorological stations or special wind surveys. At the majority of meteorological stations the indications of instruments measuring wind velocity are registered six times daily, each 4 hours. These indications provide the possibility for obtaining sufficiently accurate information on average periodical velocities and the discrepancies between them and data computed according to average hourly velocities do not exceed 1%.

However, macro- and microrelief of the surrounding locale and the degree to which the meteorological station is protected affect the anemometer readings, which should be taken into account during recalculation of velocities for each actual read-in, even if it located not far from the station.

Average wind velocities change essentially during the various periods of the day, and in different months and seasons. In accordance with this, the daily, monthly and seasonal spans of velocities characterizing the overall tendency of their changes

during the periods indicated also vary. These changes rate the macrostructure of the air flow.

The limiting values of wind velocities, data on its intensity and the microstructure of the flow over relatively short time intervals are important rate characteristics, whose knowledge is necessary for computation of strength of machines and designing of installations having automatic adjustment, orientation and so forth.

Pulsation of velocity, and consequently also the energy of the flow is caused by the general formation character of the wind's structure, local features and also the effect of landscaping and relief conditions. The importance of this characteristic is necessitated by the fact that frequently the decisive reason for destruction of a machine is not so much the overall level of wind velocity as the dynamics of the stream and variations in velocity over relatively short time intervals. They are evaluated by accelerations in the flow, the duration of gusts, coincidence of gusts in various points of the surface swept by the windmill, and the coefficient of gustiness of the flow K_{gus}, which is the relationship of the maximum measured velocity v_{max} to the average $\bar{v}$ over the selected time interval (usually no more than 2 min): /8

$$K_{gus} = \frac{v_{max}}{\bar{v}}. \qquad (3)$$

The study of wind velocity variations is most fruitful using a composition analysis of conformities and conjectural variability of the intensity of flow both in the selected time interval and on a limited area (space). If a sample of wind velocity records is schematically represented in one point of the graph in Fig. 1, it is possible to observe fluctuation 1, forming gusts, fluctuation 2 corresponding to microgusts of the wind and, finally, micropulsations of velocity 3 [10]. With an extended interval, records of seasonal changes of the overall level of velocity can be considered regular phenomena tied with variability of the elements of the large-scale circulation of the atmosphere, and the eddying formations of smaller scales forming wind gusts which are added to them can be considered to be incidental.

At meteorological stations, the average wind velocity is usually fixed over a time interval of no less than 2 min. In practice it is frequently necessary to have wind characteristics over several seconds, which, naturally, can be different. Evaluating the probability $F(v)$ of the fact that momentary velocity of the wind can be greater than the average measured value, we find the calculated velocity with a set norm of guarantee

$$F(v) = e^{-\left(\frac{v}{\beta}\right)^{\gamma}} \qquad (4)$$

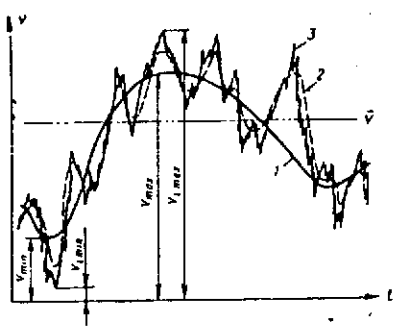

Fig. 1. Schematic illustration of wind velocity record.

where β and γ' are parameters depending on the wind rate and for meteorological stations of the seventh and eighth classes of openness are equal to 1.3-1.5 and 5-6.5 respectively [2].

The coefficient of wind gustiness is a function of a number of factors and for different geographical zones its characteristics vary within large limits. It has been established by investigation that the higher the wind velocity the lower the value of gustiness, although with this the absolute deviations of velocity from the average increase [3, 4, 28, 55]. /9

A change in K_{gus}, shown on Fig. 2, characterizes the gustiness of the stream, the area of section of which is small and equal to the area swept by the vane of the anemometer.

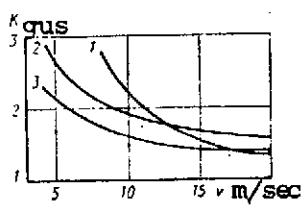

Fig. 2. Coefficients of gustiness of wind velocity: 1. according to data of G. Savitskiy; 2. according to data of V. Bol'shakov; 3. according to data of the author.

From the graph in Fig. 2 it is evident that the value of the coefficient K_{gus} is relatively high. A simultaneous record of velocity by anemometers located in a plane perpendicular to the main direction of the flow (Fig. 3), shows that velocity gusts generally have a chaotic character and do not coincide either in value or in direction. Therefore, the total intensity of the gusts with respect to area decreases by comparison with the gustiness of a flow having small cross section. The reduction in gust intensity becomes stronger as the section area grows larger. This is also evident from Fig. 4, where the change in velocity is recorded using low-inertia (braked) and normal anemometers, placed at a distance of 10 m from each other. The daily span of wind velocity at points located at a distance of several kilometers from each other shows that even the average hourly velocities differ essentially from each other. /10

The average intensity (power of gusts over time interval T can be evaluated with the expression

$$v^2(t) = \lim_{T \to \infty} \int_0^T [\bar{v}(t)]^2 \, dt, \qquad (5)$$

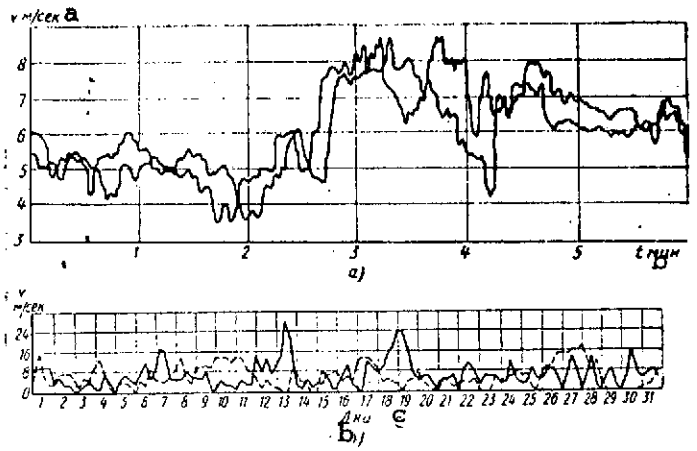

Fig. 3. Simultaneous record of wind velocity by two anemometers: a. located at a distance of 20 m apart; b. average span of velocity at points separated by several kilometers.

Key: a. m/sec; b. min; c. days.

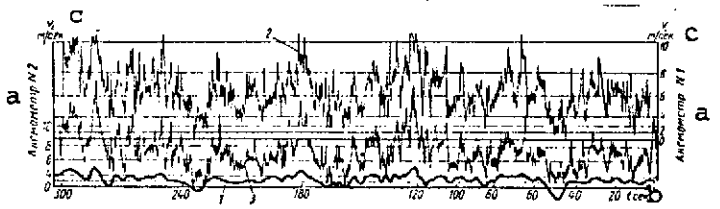

Fig. 4. Simultaneous record of wind velocity: 1. electroanemometer; 2. low-inertia anemometer No. 1; 3. low-inertia anemomenter No. 2.

Key: a. Anemometer No. ...; b. sec; c. m/sec

the type

$$v = v_1 \left(\frac{h}{h_1}\right)^{\frac{1}{n}} \qquad (6)$$

or

$$v = v_1 \frac{\lg \frac{h}{h_0}}{\lg \frac{h_1}{h_0}}, \qquad (7)$$

where $\bar{v}$ is the velocity of the flow during the gust, averaged at the given moment in time according to the measurements of several anemometers.

We will examine characteristics allowing judgment of changes in wind velocity according to the height of the surface layer with a thickness of up to 300 m; we will evaluate the vertical profile of the wind. The effect of the earth's surface on the velocity and direction of air streams decreases as the distance from the earth decreases. Therefore, the velocity of the wind at great heights usually increases, and its gustiness and acceleration in stream decrease. The velocity gradient in summer is, as a rule, less than in winter, which is first of all explained by the different character of the change in temperature jump during the winter and summer periods.

The results of direct observations in many regions at different heights showed that with an adiabatic condition of the atmosphere, the vertical profile of the wind $v(h)$ is well approximated with dependencies of

where v is the sought velocity of the wind at height h; v_1 is the velocity of the wind measured near the earth at height h_1, h_0 is the height at which the velocity of the wind is equal to zero. /11

Height h_0 is usually seen as the measure of roughness of the underlying surface and is taken as equal to: 0.5 cm for a snow covering; 3.2 cm for a surface covered by low grass; and 5-7 cm for a surface with higher growth. The maximum value of h_0 = 20 cm.

The direction of the wind usually plays a smaller role in utilizing wind energy. However, in various landscape conditions, winds of different directions have characteristic features: higher or lower gustiness and velocity. Their recurrence is determined according to the wind rose which shows in graphic form what percent of the total time of a year the wind has a certain direction. The angular gradients of velocity show the essential influence on the operation of automatically orienting mechanisms and on the value of gyroscopic loads; gustiness caused by the turbulent structure of the stream has an effect on the operation of systems having automatic adjustment of rotation frequency and windmill power limiting, and also affects the reliability of the machinery.

The energy of the air flow E, having a cross-sectional area of F, is determined according to the expression

$$E = \frac{mv^2}{2} \text{ J.}$$

Since the mass of air m flowing with a velocity of v through this section in a second is

$$m = \rho F v \text{ kg/sec,} \qquad (8)$$

the energy is

$$E = \frac{\rho v^3 F}{2} \text{ J/sec,}$$

where ρ is the air density, equal under normal conditions to 1.23 kg/m^2 (at t = 15°C and p = 760 mm Hg).

Therefore, the energy of the wind changes proportional to the cube of its velocity. The windmill can transform into useful work part of its energy, which is evaluated with the coefficient of wind energy utilization ξ.

§ 2. The Wind Energy Cadaster

The wind energy cadaster is understood to mean the coincidence of accessible and necessary information characterizing the wind as a source of energy and allowing the learning of its energy-producing value. It is a system of numerical characteristics of wind rate in various zones, on the basis of which it is possible to judge the rates and duration of the work of a machine of a certain power, the total development of energy and others. The most important cadastral characteristics are repetition (density of distribution) of various velocities, alternation of working and calm periods and rates of maximum (storm) velocities. The values for average annual v_a and average seasonal v_s velocites are also important and, primarily, are convenient cadastral characteristics of the overall level of wind intensity, but their amounts still do not determine in full measure the effectiveness of using wind machines. /12

The function of statistical regularity of the frequency of variations of wind velocities over a definite time should be considered the most important characteristics. Knowing the regularity, determining the type and parameters of this function, and having the characteristics of the wind energy-producing machine, it is possible to calculate its development, duration of stoppages, coefficient of utilization of a set power, economic effectiveness, etc.

Utilization of the curves of repetition of M. S. Pomortsev and G. Gullen, constructed for propositions of normal distribution of velocities for all regions and zones cannot be accepted as sufficiently accurate [11]. They can be used only in actual and partial cases, although on the whole they can be accepted as satisfactory for many regions of the country.

G. A. Grinevich [11] proposed the following equation for a distribution curve of essentially positive values for a number of zones:

$$y = \alpha_1 x^p e^{-kx^n} \tag{10}$$

(where y, x are the variables; α_1, p, k , n are the coefficients depending on the type of curve) on the basis of which, transferring to absolute values of velocity for the wind and their repetition in the parts from the times of the period, we obtain

$$t = \frac{\Delta v}{\bar{v}} \alpha_1 \left(\frac{v}{\bar{v}}\right)^p e^{-k\left(\frac{v}{\bar{v}}\right)^n}, \tag{11}$$

where t is the frequency or repetition of velocity v; $\bar{v}$ is the average velocity over the calculated period; v is the velocity

whose repetition is determined in the interval from ($v - \frac{\Delta v}{2}$) up to ($v + \frac{\Delta v}{2}$); Δv is the selected intervals of velocity gradiation.

Due to the typification of wind velocity repetition rates for the majority of the regions of Central Asia and Kazakhstan, the coefficients of the curve and the basic statistical characteristics of distribution were determined, allowing provision in tabular or graphical form of numerical materials according to the number of hours of activity at various velocities which were convenient for practical usage.

For some zones with relatively high values of $\bar{v}_a$ and $\bar{v}_s$ (6-7 m/sec), the results of wind rate typification, which well satisfy the practical demands, are obtained by using dependencies emanating from an equation of R. D. Gudrich, in the differential form. /13

$$y = knx^{n-1}e^{-kx^n}. \tag{12}$$

This equation was taken by M. V. Kolodin for equalizing empirical distributions of velocities [16]. Knowing the parameters a and k of equation

$$t_{v_i} = \frac{\Delta v}{v_i}\left(\frac{v_i}{\bar{v}}\right)^{a-1} e^{-k\left(\frac{v_i}{\bar{v}}\right)^a}, \tag{13}$$

it is easy to compute how many hours during the year (season) the wind with the velocity sought will be repeated. These data are usually introduced in the form of tables and graphs. As a rule, repetition in them is related to the height at which the wind vane is set (usually, 12-15 m).

The height h_{wm} of the distance between the center of the windmill and the ground can be any. To determine the repetition of velocity at height h_{wm}, one should use the conversion formula

$$t_{v_{h_{wm}}} = t_{v_{15}} \frac{h_{wm}}{h_{15}} = k_{t_{h_{wm}}} t_{v_{15}} \cdot \text{hours}, \tag{14}$$

where $t_{v_{15}}$ is the repetition of velocity v at a height of 15 m; $k_{t_{h_{wm}}}$ is the coefficient of conversion of wind velocity repetition, whose values, depending on the calculated height are the following:

h_{wm} in m	15	10	7	5	4
$k_{t_{h_{wm}}}$	1.0	0.933	0.875	0.82	0.785

Therefore, it turns out that with other heights the original repetition relates somehow to the new values for velocity. Summary data of conversion of velocity repetition into parts per thousand (0/00) are presented in Table 1 for two types of velocity distributions defined by the expression:

With wind velocities $\bar{v}$ = 3-5 m/sec

$$t = 1258 \frac{\Delta v}{\bar{v}} \left(\frac{v_i}{\bar{v}}\right)^{0.45} e^{-0.867 \left(\frac{v_i}{\bar{v}}\right)^{1.45}}; \quad (15)$$

With $\bar{v}$ = 6-7 m/sec

$$t = 1571 \frac{\Delta v}{\bar{v}} \cdot \frac{v_i}{\bar{v}} e^{-0.785 \left(\frac{v_i}{\bar{v}}\right)^2}. \quad (16)$$

The repetition of high wind velocities is evaluated with a less accurate distribution curve. L. Ye. Anapol'skaya and L. S. Gandin [2] proposed definition of the value n, the indicator, from the number of years (orfrom what part of the year) one incidence of a velocity higher than v was observed, according to the expression /14

$$n = \frac{1}{N_v F(v)}, \quad (17)$$

where N_v is the number of observations over the course of the year.

Since the differential function of probability of the chance value f(u) is connected with the integral function F(v) by the expression

$$f(u) = -\frac{dF(v)}{dv}, \quad (18)$$

then

$$f(u) = \frac{\gamma'}{\beta} \left(\frac{v}{\beta}\right)^{\gamma'-1} e^{-\left(\frac{v}{\beta}\right)^{\gamma'}}, \quad (19)$$

where γ' and β are the parameters of the distribution curve.

Over the course of a year, identical wind velocities, including maximum ones, are repeated extremely unevenly. Changes in the mathematical expectation of $\bar{v}_{mo}$ according to the data of some meteorological stations (Table 2) shows the presence of a summer minimum of wind rate intensity, which frequently coincides with an increased requirement for energy. This, in a number of zones, increases the wind energy deficit. /15

TABLE 1. REPETITION OF WIND VELOCITIES AT VARIOUS HEIGHTS FOR TWO TYPES OF DISTRIBUTIONS

v_i in m/sec	Values of $\bar{v}_a$ and $\bar{v}_s$ at a height of 15 m in m/sec																								
	3					4						5				6					7				
	Repetition of velocities in 0/00 from T at height h in m																								
	15	10	7	5	4	15	10	7	5	4	15	10	7	5	4	15	10	7	5	4	15	10	7	5	4
0.5	56	61	68	76	80	37	40	43	48	52	12	13	14	15	15	11	11	12	13	13	9	10	11	11	11
1	214	232	253	271	282	149	165	178	192	203	60	68	78	82	99	42	48	55	62	68	31	36	41	41	50
2	216	226	238	245	251	168	178	188	199	208	110	124	137	142	166	79	89	101	114	122	60	69	77	85	92
3	176	178	176	173	171	156	162	167	171	174	141	155	167	171	195	107	119	131	145	155	82	94	104	114	123
4	128	124	116	109	104	132	133	133	131	130	152	161	169	172	180	122	133	143	156	161	98	109	120	128	133
5	86	79	70	63	58	105	103	99	94	89	143	146	148	147	140	125	133	138	145	146	106	119	123	133	134
6	53	47	40	32	28	79	74	69	63	58	121	118	112	110	96	119	122	122	120	118	107	114	118	121	123
7	32	25	20	16	13	58	52	47	42	37	93	85	77	73	56	105	103	99	90	85	101	103	105	104	102
8	18	14	10	8	6	40	34	30	26	21	67	58	47	43	30	86	81	74	64	57	92	90	87	83	75
9	10	7	5	3	3	28	24	18	14	12	43	34	27	24	13	66	58	50	40	34	78	73	67	62	55
10	5	3	2	2	2	18	14	12	8,5	7	27	19	13	11	6	48	40	32	24	20	63	57	50	43	38
11	2	2	1	1	2	11	8	6	4,5	3,5	15	10	6	5,5	2,5	33	26	19	13	10	51	42	35	28	24
12	2	1	1	1		7	4	4	2,5	2,5	8	4,5	3	2,5	1	22	16	12	7	5,5	38	30	23	18	15
13	1	1				4	3	2	2	1,5	4	2	1	1	0.5	14	9	5	3,5	2,5	27	20	15	12	8
14	1					3	2	2	1	1	2	1,5	1	1		8	5	3	1,5	1	19	13	9	6	4,5
15						2	2	1	1	0.5	1	1				5	3	2	1	1	13	8	5,5	4	2,5
16						2	1	1	0.5		1					3	2	1	1		8	6	3,5	2,5	1,5
17						1	1									2	1	1			6	3	2,5	1,5	1,0
18																1	1				4	3	1,5	1	0.5
19																1					3	2	1		
20 and higher											←					1					3,1	2	1		

TABLE 2. WIND RATE AT SOME METEOROLOGICAL STATIONS

Name of station	Years of observation	$\bar{v}_{mo}$ in m/sec												$\bar{v}_s$ in m/sec				$\bar{v}_a$ in m/sec
		I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII	I–III	IV–VI	VII–IX	X–XII	
Balkhash	1958—1966	5.3	7.0	5.9	5.9	5.6	5.3	5.3	5.2	5.3	4.8	4.7	5.0	6.1	5.6	5.3	4.8	5.4
Rostov-on-the-Don	1958—1960	6.5	4.6	5.3	5.3	4.8	4.7	5.6	3.9	4.1	5.4	9.8	6.0	5.5	4.9	4.5	7.3	5.3
Tselina (Rostovskaya oblast')	1958—1963	4.0	4.2	4.1	4.0	3.5	3.1	2.5	3.1	2.7	3.7	5.0	4.0	4.0	3.5	3.0	4.2	3.7
Volgograd-Gumrak	1958—1963	6.1	6.5	6.3	6.2	6.0	6.3	4.8	5.2	5.5	6.0	6.4	6.4	6.3	6.2	5.2	6.3	6.0
Baku	1958—1965	5.5	5.9	6.1	5.8	5.5	5.7	6.7	5.9	5.9	5.8	5.0	4.7	5.8	5.6	6.2	5.2	5.7
Kolchugina (Kemerovskaya oblast')	1958—1961	4.5	5.0	4.2	4.3	4.1	3.6	2.8	2.9	3.5	5.4	4.9	4.3	4.6	4.0	3.0	4.9	4.2
Mugodzharskaya	1958—1963	8.1	8.0	6.7	5.6	5.0	4.3	4.0	4.2	4.3	5.7	6.2	8.1	7.1	5.0	4.2	6.7	5.7
Makhachkala	1958—1965	5.1	5.5	5.1	5.5	5.3	4.1	4.3	5.1	4.8	5.0	4.7	5.0	5.2	5.0	4.7	4.9	4.9
Gur'yev	1958—1964	6.3	6.0	5.9	6.3	6.3	5.2	4.6	4.3	4.8	5.3	5.4	5.0	6.0	6.0	4.6	5.2	5.5
Kuybyshev	1958—1962	4.0	4.1	4.5	3.9	3.9	4.0	2.8	3.0	3.5	3.9	3.9	4.4	4.2	4.0	3.0	4.0	3.7
Dzhungariya (Alma-Ata)	1958—1960	11.3	10.0	7.9	5.0	4.3	2.4	2.6	3.2	5.0	6.5	8.0	9.5	9.7	3.9	3.6	8.0	6.1
Fort Shevchenka	1958—1964	7.4	7.2	6.8	6.3	6.4	5.2	5.7	5.2	6.2	6.5	7.7	7.7	7.1	6.0	5.7	7.3	6.5
Gigilgay (Azerbaydzhan SSR)	1952—1962	5.0	4.7	5.2	5.6	4.7	4.7	5.2	5.4	5.2	4.7	4.8	4.6	5.0	5.0	5.3	4.7	5.1
Mashtagi (Azerbaydzhan SSR)	1952—1962	6.9	6.4	7.0	6.3	6.2	6.0	6.9	6.1	5.4	5.6	5.7	5.4	6.8	6.2	6.1	5.6	6.1
Turgay	1958—1964	4.6	4.5	4.9	4.7	3.8	4.4	4.5	4.1	4.0	4.4	4.2	4.0	4.7	4.3	4.2	4.2	4.3
Odessa	1958—1966	5.5	5.3	5.2	5.3	5.3	4.1	4.0	4.2	4.1	4.3	5.0	5.1	5.3	4.9	4.1	4.8	4.7

An increase in the overall level of wind rate intensity does not always indicate a reduction in the maximum direction of energy-producing calms t_{calm}, which define the nonworking periods (idle periods) of the machine due to meteorological conditions. Idle periods are in turn connected with minimum working velocities v_0 of the machine, which are measured within limits of from 3 to 5 m/sec, depending on the type of machine. As is evident from Fig. 5, t_{calm} is a function of v_0. Only 5-7% of calms have a duration of more than 2 days, and the limiting duration of t_{calm} = = 36-96 h. Thus, in Kazakhstan, in zones with v_a = 5.5-5.7 m/sec, the values of the maximum nonworking periods of the machine change /17 depending on v_0 within the following limits: with an increase in v_0 by 1 m/sec (from 3 to 4 m/sec) the minimum duration of the energy-producing calm increases by 7-20%, and with Δv_0 = 2 m/sec, it increases by 45-60% (Table 3). In different years, the repe- /18 tition of short duration calms observer over the course of the same month varies, as a rule, within a relatively large range, and the limiting values change less.

According to materials of many years' observations, it is possible to trace some, not strongly expressed tendencies toward a decrease in the total number of calms during the winter period. Concerning duration of uninterrupted calms, in winter their probability increases.

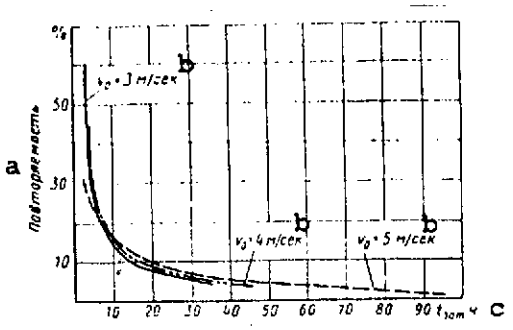

Fig. 5. Mathematical expectation of duration of energy-producing calms depending on values of initial working wind velocities.

Key: a. Repetition; b. m/sec; c. t_{calm}, h

Integral curves of duration of calms of the type shown in Fig. 6 provide the possibility for making a preliminary evaluation of the limits of capacities for wind energy storage cells, on which one should orient oneself. Thus, in June, according to data from the Mugodzharskaya station, the limiting duration of a calm was 90 h, but it was repeated once per 4 years. If one orients oneself on providing energy for 95% of the calms, the uninterrupted duration of the calculated period of calms is 42 h, and with provision for 90% of the calms, it is 33 h. Naturally, one should not strive toward accumulation of energy for the periods of all, even the longest but rarely repeated calms, since this will lead to unjustified high capital expenses on storage cells and, over the course of a year, a low degree of its useful capacity. We note that since the number of observations of the chronological wind velocity rates is relatively small, the data presented illustrates only the overall tendency of repetition of calms for the zone.

TABLE 3. MATHEMATICAL EXPECTATION OF LIMITING DURATIONS OF UNINTERRUPTED ENERGY-PRODUCING CALMS IN HOURS (DATA OF A 10-YEAR SERIES OF OBSERVATIONS)

Months	Meteorological station					
	Balkash	Mugodzharskaya	Balkash	Mugodzharskaya	Balkash	Mugodzharskaya
	v_0					
	3		4		5	
January	30	72	42	84	42	126
February	24	42	33	57	51	78
March	63	57	66	57	69	72
April	18	21	33	39	78	51
May	15	69	39	87	57	90
June	18	24	39	36	39	90
July	18	24	24	42	45	60
August	18	—	39	—	42	—
September	18	39	27	51	39	60
October	18	39	48	45	57	54
November	60	57	57	69	93	114
December	33	81	75	87	105	105
Maximum for a year	63	81	75	87	105	126

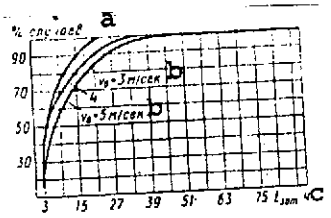

Fig. 6. Integral curves of duration of calms at v_a = 5.5-5.7 m/sec.

Key: a. Percent of incidents; b. m/sec; c. t_{calm}, h

The mathematical expectations of limiting durations of calms (Table 4) can be used during preliminary evaluation of storage cells for the different norm of required energy provision.

The maximum values of t_{calm} depend on the physical and geographical conditions of the locale. Thus, for the Betpak-Dala desert, t_{calm} less than 3 days is observed annually, up to 5 days is observed once in 5-10 years, and over 5 days is practically never repeated [25]. The uninterrupted duration of periods with active wind velocities is usually less than t_{calm} (Table 5).

Data on wind characteristics usually have an integral character, and since practice presents ever increasing requirements for establishment of conformities in the chronological variability of wind rates, the task of modeling these rates arises. Since in modeling processes lasting up to a year must be examined, it is necessary to conduct a large number of identical observations. This is connected with the fact that the range of wind velocities differs during the different seasons and it is desirable to study the genesis of the process and examine the inner ties of $v(t)$ with the solar activity, the condition of the atmosphere and other factors.

In wind energetics, mathematical modeling has an important meaning for evaluating capacity of storage cells and power of reserving installations, and also for diagrams of energy output by the machine, since wind energy cannot be stored until such time as useful work is accomplished by it. In other words, a wind installation without a storage cell can work only according to an uncontrollable diagram.

The rate of chronological variability of velocity is a complex regularly conjectural process. Since the mathematical theory of such processes is insufficiently developed, one must resort to a number of simplifications. This is even more necessary since the process $v(t)$ over an extended period cannot be considered stationary, while at the same time for minutes it is usually regarded as stationary, whose characteristics are expressed with the first two moments of the distribution -- the mode and correlation function.

According to data of L. B. Gartsman, for evaluating v_{max}, it is possible to use function [10]

$$F_1 = \alpha_2 \left[\frac{v}{M(v)}\right]^p e^{-k_1\left[\frac{v}{M(v)}\right]^n}, \qquad (20)$$

TABLE 4. MATHEMATICAL EXPECTATION OF THE LIMITING DURATION OF CALMS IN HOURS

Norms of provision in %	Month	Meteorological station								
		Balkash			Mugodzhar-skaya			Turgay		
		Level of velocity producing calm in m/sec								
		3	4	5	3	4	5	3	4	5
95	I	27	36	45	39	79	84	41	63	68
	II	21	26	33	23	36	36	37	51	70
	III	21	22	33	49	42	60	48	62	72
	IV	20	27	26	20	26	45	26	39	51
	V	12	20	26	22	32	51	22	33	47
	VI	13	27	30	21	32	72	29	29	37
	VII	15	19	38	21	27	45	27	36	52
	VIII	12	21	27	35	36	51	22	36	49
	IX	16	22	28	—	39	51	33	41	60
	X	27	57	60	29	39	47	29	43	45
	XI	17	33	39	48	80	87	38	47	70
	XII	21	33	51	51	78	102	63	101	120
	$t_{calm\ max}$	27	57	60	51	79	102	63	101	120
	t_{calm}	18,5	29.5	36.3	29.9	45.5	60.9	34.6	48.4	61.7
90	I	23	30	39	34	63	65	33	53	57
	II	15	21	27	16	28	26	32	39	54
	III	17	19	28	42	39	42	35	47	63
	IV	18	24	21	17	21	42	22	30	43
	V	9	15	20	18	20	41	18	24	36
	VI	9	21	25	18	24	39	23	23	27
	VII	12	16	30	18	23	37	23	28	36
	VIII	8	17	21	28	29	39	19	29	41
	IX	14	18	23	—	30	42	27	31	48
	X	24	39	49	24	35	39	24	36	38
	XI	14	27	32	41	65	67	30	36	58
	XII	17	27	38	39	64	93	51	84	114
	$t_{calm\ max}$	24	39	49	42	65	93	51	84	114
	t_{calm}	15.0	22.8	29.4	24.6	36.8	47.7	28.1	38.4	51.2
80	I	18	21	28	29	32	36	22	26	33
	II	10	16	20	11	15	16	21	28	36
	III	12	14	22	27	33	34	18	29	56
	IV	15	17	19	14	17	39	15	21	27
	V	6	10	14	15	16	29	13	17	25
	VI	6	13	18	14	21	24	16	16	17
	VII	9	12	18	14	17	27	17	20	21
	VIII	5	12	15	19	20	28	14	22	32
	IX	9	14	17	—	20	30	18	23	33
	X	18	22	33	17	27	30	17	25	27
	XI	11	18	24	33	45	48	20	26	42
	XII	12	18	28	23	51	75	26	60	66
	$t_{calm\ max}$	18	22	33	33	51	75	26	60	66
	t_{calm}	10,9	15,6	21,3	18	26,2	34,4	17,9	26,1	34,6

TABLE 5. UNINTERRUPTED DURATION OF ACTIVE ENERGY-PRODUCING PERIODS IN HOURS

(a) Среднепериодная скорость ветра в м/сек	(b) Минимальный уровень рабочей скорости периода в м/сек			
	3	4	5	6
3	52	34	20	12
4	65	50	36	25
5	73	60	47	37
6	77	66	55	45

Key: (a) Periodic wind velocity in m/sec
(b) Minimum level of working velocity of the period in m/sec

where p is the provision; α_2, k_2, n are the statistical characteristics of the process; $M(v)$ is the function of the average value of velocity.

Examination (together with M. A. Bondarevskaya) of a number of characteristics of chronological variability of the wind over months and a year allowed development of some regularities which are evident from the span of wind velocities over the January-March period, averaged over 10 months, according to data from the Astrakhani meteorological station (Fig. 7) and from graphs of the daily span of velocities over 1959-1960, determined according to four sections of random functions corresponding to the times of observation (Fig. 8). It was established that in this zone the velocity v and its quadratic deviation σ increased to 13 hours, while at the same time, σ/m varies within limited boundaries. This gives basis for proposing the possibility of interpreting the process $v(t)$ as a coincidence of several factors, one of which has a periodic character. The mathematical expectation m_{v_1} is an average velocity

$$m_{v_i} = \frac{\sum_{i=1}^{n} v_i}{n}, \tag{21}$$

but dispersion, i.e. scattering of velocities relative to the average value is

$$D = \frac{\sum_{i=1}^{n} (v_i - m_{v_i})^2}{n-1}. \tag{22}$$

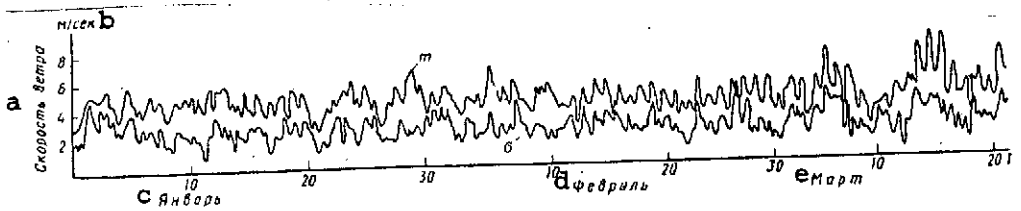

Fig. 7. Average span of wind velocities according to data of the Astrakhani meteorological station over 1939-1948: m -- mathematical expectation, σ -- quadrative deviation.

Key: a. Wind velocity; b. m/sec; c. January; d. February; e. March

Examination of the relationships of preceding and succeeding values of velocity for the same meteorological station, divided into sequentially increasing time intervals showed that the coefficient of correlation /22

$$r = \frac{\sum_{i=1}^{n-m} [(v_i - m_{v_i})(v_{i+t} - m_{v_{i+t}})]}{(n-m)\,\sigma_{v_i}\sigma_{v_{i+t}}} \qquad (23)$$

is 0.6-0.65 for t = 6 h and 0.22 for t = 18 h. For certain months, greater r (up to 0.76) were observed during the daylight hours. The correlation is almost totally damped out over the course of the day (t = 24 h).

Similar data can be used for constructing correct mathematical and graphical approximations of the process $v(t)$, considering superimposition of occasional variations onto the regular tendency of the chronological span of velocity. Naturally, for definite groups of curves $v(t)$, typified beforehand for different zones according to corresponding indicators, development of their own models is required. /23

Over a year or season the process $v(t)$ is not stationary, but its deviation from the average value can be stationary and then it is possible to see it as a Markovian process. With this, over an extended period, commensurate with the cycle of energy output adjustment, it is possible to write

$$v(t) = f(t) + q(t)\,\xi_v(t), \qquad (24)$$

where f(t) is the climatological span of velocity v, caused by the annual rotation of the earth around the sun and seasonal changes in circulation of the atmosphere; q(t) is the daily span of velocity caused by changes in the condition of the turbulent atmosphere; $\xi_v(t)$ is the fluctuation of the daily span caused by the turbulent structure of the flow.

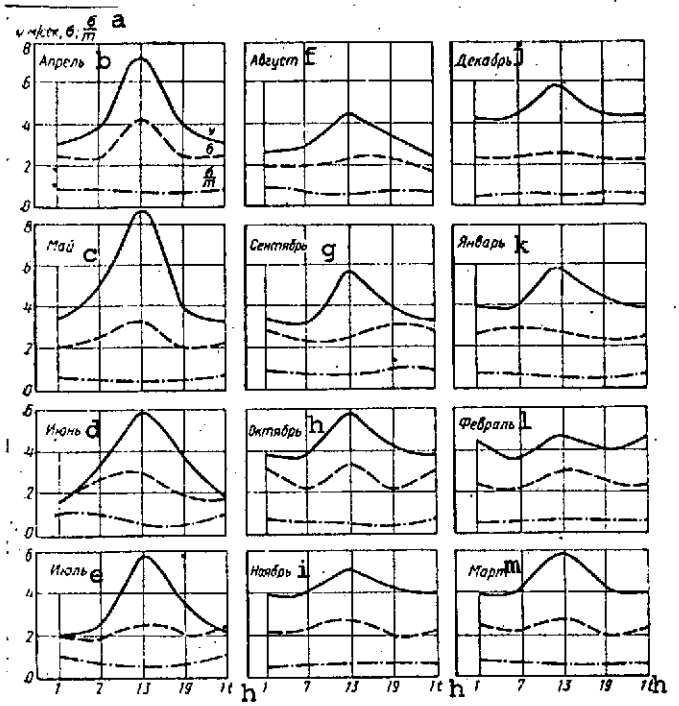

Fig. 8. Changes in daily wind velocities according to data of the Astrakhani meteorological station over 1959-1960.

Key: a. v, m/sec, σ, σ/m; b. April; c. May, d. June; e. July; f. August; g. September; h. October; i. November; j. December; k. January; l. February; m. March

It is possible to point to several courses for solving the problem. The first is to regard f(t) and q(t) as nonrandom functions (processes) and approximate them with mathematical expressions, and regard $\xi_v(t)$ as a random process for which numerical characteristics are defined; the second course is to find spectral characteristics for different zones and type of curve according to their Grenader-Rozenblat method [expression (25)]; the third course is in simplified form establishing constant coefficients [according to expression (26)], looking not at the fluctuation in daily velocity span, but at deviations from the averaged daily span, which in the first approximation can be considered stationary:

$$v(t) = \sum_{n=1}^{n} (a_m \cos \lambda_m t + b_m \sin \lambda_m t) + \xi_v(t); \quad (25)$$

$$v(t) = C_1 (1 + \sin \omega_v t) + C_2 \sin \omega_v t + \xi_v(t), \quad (26)$$

where a_m, b_m, C_1 and C_2 are coefficients of the curve (amplitude of the process); λ_m is the phase of the process; ω_v is the frequency of the process.

§ 3. Wind Energy-Producing Resources

Data is often presented in the literature about wind energy which can be obtained from 1 km^2 of the earth's surface. It can therefore be calculated that the established power of wind energy-producing machines across the USSR can comprise nearly 11 billion kW with an annual energy development of more than $1.8 \cdot 10^{13}$ kW hours [18]. During recent years it is fair to point out that the /24

specifics of wind as an energy-producing source, the character and technology of its utilization make similar calculations and the numbers presented abstract to a certain degree, since practical realization of all this energy and its transformation into mechanical, thermal or electrical energy is impossible, and for the time being only a very small part of it, measured in hundredths of a percent is being successfully profitably utilized.

The potential possibilities for using wind energy are practically not limited in the majority of regions. However, these possibilities constantly change depending on the perfection of technical means, using energy from those heights to which it is possible to penetrate with new machines, on changes in the structure of the overall power production balance of the country, economic criteria, etc.

For instance, in the end of the 1950's, projects appear for altitude (tropopause) wind electrical stations, whose use could essentially increase the proportion of practical utilization of wind energy. In the meantime, resources such as these remain, naturally, unchanged. Therefore, calculations for wind energy resources which provide objective prognosis must be based only on actual engineering solutions, designed for definite purposes, considering the thickness of the air layer which will be used by the motors (it is determined by the diameter of the windmill), and its location relative to the surface of the earth. Naturally, the number of machines installed on definite areas must be computed beginning not only with the physical possibilities, dictated by aerodynamic considerations and the inadmissibility of harmful mutual effect of windmills, but also must be selected with consideration for location of the consumers, the relief of the locale and other conditions.

At the same time, the qualitative side of evaluating wind energy resource depending on local conditions can be exposed if it is judged according to the value for average annual wind velocity. Domestic and foreign experience shows that utilization of low power (5-6 kW) wind energy-producing machines is almost always economically justified in regions where $\bar{v}_a \geq 4$ m/sec, and use of modern wind-electrical stations with a high power (30 kW and higher) is justified with $\bar{v}_a \geq 5.5$-6 m/sec. The construction of tall wind electric stations will doubtless expand these regions.

Chapter 2. Brief Information from Aerodynamics

Aerodynamics studies the motion of air and the interaction between the air and a solid body moving in it. Since the relative motion is examined, it is applicable to wind energy-producing machines to study the mutual action of the windmill and other elements of the wind engine, which is itself stationary, with the /25

air stream, which has a definite velocity and direction. Usually the wind stream, even in a relatively small section, is not uniform; its velocity in various sections is different and velocity gusts are mixed in phase relative to each other.

Aerodynamics of the wind engine are studied in sections of industrial aerodynamics. While theoretical aerodynamics is by its character basically applied mathematics and mechanics and in examination of many phenomena it is often based on their mathematical, electrical and other models, experimental aerodynamics studies the phenomenon itself, investigates real objects or their physical models, generalizes the results of experiments and defines experimental coefficients.

§ Basic Laws and Concepts of Aerodynamics

Aerodynamics is founded on several basic laws. In accordance with the first law -- the law of inverse motion -- it is possible to examine a stationary body surrounded by a moving medium, or, conversely, the motion of a body in a stationary medium, with which the results of examination will be identical. The second law -- the law of constancy of mass, expressed with an equation of continuity, allows determination of the velocity of a liquid in various sections of an enclosed space. With this, liquid is understood to mean a medium having both a gaseoform state and one sufficiently liquid to form drops. At low velocities the motion of gas and air can be looked at according to laws of hydrodynamics. Examining the elementary stream of liquid AB under conditions of established motion, we assume that over time Δt liquid mass M_1 passes through section A-A (Fig. 9, a), and mass M_2 flows through section B-B.

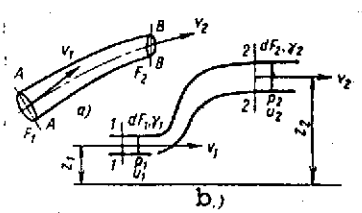

Fig. 9. Toward derivation of the equations: a -- of continuity; b -- Bernoulli's.

Since the motion is established, and the liquid cannot flow through the lateral walls due to the fact that its velocity is directed along tangents to the walls throughout, then

$$M_1 = M_2 = M = \text{const} \tag{27}$$

or

$$\rho_1 F_1 v_1 = \rho_2 F_2 v_2 = \rho v F = \text{const}. \tag{28}$$

For an incompressible liquid $\rho_1 = \rho_2 = \rho$, therefore

$$F_1 v_1 = F_2 v_2 = F v = \text{const}. \tag{29}$$

Bernoulli's equation is a reflection of the third law -- the law of conservation of energy, which is applied to the stream of liquid in established motion. /29

We will examine the second energetic components of the stream flowing in through section 1-1 and out through section 2-2 (Fig. 9, b). The potential energy of the liquid is equal to the product $dF_1 v_1 \rho_1 g$, mutliplied by the leveling height z_1.

The energy caused by the pressure of liquid particles is equal to the product of pressure p_1 by the area of the stream section dF_1 and the velocity of liquid flow v_1, i.e. $p_1 dF_1 v_1$.

The kinetic energy is equal to

$$\frac{m_{sec} v_1^2}{2} = \frac{\rho_1}{2} dF_1 v_1 v_1^2 = \frac{\rho_1 dF_1 v_1^3}{2} . \qquad (30)$$

Finally, the internal thermal energy of 1 kg of liquid passing through section 1-1 will be U/A, where U is the internal thermal energy; A = 427 kcal/kg·m is the mechanical equivalent of heat.

The total energy is sections 1-1 and 2-2 will remain constant. Expressing the balance of specific energy through the balance of pressures, with consideration for the loss in pressure h_n between sections 1-1 and 2-2, we obtain

$$z_1 + \frac{p_1}{\rho_1 g} + \frac{v_1^2}{2g} + \frac{U_1}{A} = z_2 + \frac{p_2}{\rho_2 g} + \frac{v_2^2}{2g} + \frac{U_2}{A} + \Sigma h_n . \qquad (31)$$

For an incompressible liquid $\rho_1 = \rho_1 = \rho$; $U_1 = U_2 = U$,

$$z_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} = z_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + \Sigma h_n , \qquad (32)$$

where $p/\rho g$ is the piezometric pressure; $v^2/2g$ is the velocity pressure; Σh_n is the total loss in pressure on the path between the selected sections.

Cavitation of liquid. Cavitation is expressed in the appearance of interruptions in the flowing medium caused by a reduction in pressure to the zero value, and occurs in cases where the potential energy of the medium is transformed into kinetic and velocity v becomes maximum. For air, the velocity at which cavitation begins is

$$v_{cav} = v_{max} = \sqrt{2\frac{k}{k-1}\frac{p_0}{\gamma_0}}, \qquad (33)$$

where k = 1.41 is the adiabatic index.

For water, the limiting value in a pipe is /27

$$v_{cav} = \sqrt{2g\left(z_1 + \frac{p_0}{\gamma}\right)}. \qquad (34)$$

Flow and circulation of velocity. If on a diagram of flowing ideal liquid curve OABC (Fig. 10) is drawn, at any point A the liquid will have some velocity v. Designating section OA through S and considering the value of S positive in the direction of the arrow, we isolate element dS and project velocity v in the direction of the tangent at point A. Multiplying the projection of velocity v_S by dS is called tracing velocity dG along the element. Summarizing the elementary flows of velocity along line AOBC, we have

$$G_{OC} = \int_{OC} v_s dS \qquad m^2/sec \qquad (35)$$

With an enclosed contour (Fig. 11) we have circulation of velocity

$$G = \oint v_s dS = \oint v \cos\alpha \, dS. \qquad (36)$$

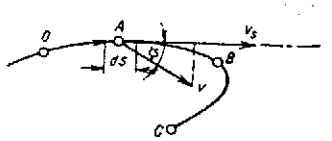

Fig. 10. Flow of velocity along a contour.

Potential of velocity. The function of space $f(x,y,z)$ whose partial derivative in any direction gives a projection of v in this direction, is called the potential of velocity.

$$v_s = \frac{\partial f}{\partial S}. \qquad (37)$$

A stream having the potential of velocities is called potential. A stream with a single-valued potential of velocities has a zero value of velocity circulation along the enclosed contour.

Differing from an ideal liquid, real liquid has viscosity, which causes forces of internal friction. At the very surface of

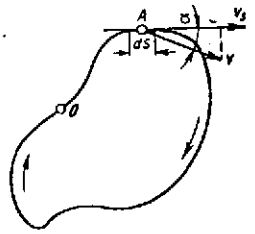

Fig. 11. Circulation of velocity.

the body, the velocity of the liquid is equal to zero, but at a very small distance the velocity approximates velocity v_n of the potential stream. This thin layer, where v increases from 0 up to $0.99v_n$, and the fluid is strongly whirled, is called the boundary layer. While dv/dy is the name for the gradient of velocity according to normals up to the surface, the tangential force caused by viscosity will be /28

$$dX = \mu \frac{dv}{dy} dS, \tag{38}$$

where μ is the coefficient of dynamic viscosity in $n \cdot sec/m^2$; dS is the value of area in m^2.

Having isolated μ for density, we obtain the coefficient of kinematic viscosity

$$\nu = \frac{\mu}{\rho} \ m^2/sec. \tag{39}$$

For normal conditions ($t_0 = 15°C$, $p_0 = 760$ mm Hg), $\mu_0 \simeq 1.9 \cdot 10^{-5}$ $n \cdot sec/m^2$ and $\nu_0 = 1.45 \cdot 10^{-5}$ m^2/sec.

With an air stream flowing past a body, the layer of air in the immediate proximity to the surface of the body is braked and whirled. This is not difficult to prove, since the circulation of velocity along a contour is not equal to zero. Diagrammatically the stream can be seen as consisting of three zones (Fig. 12, a): I -- potential area, where there is almost no difference in velocities of the layers; II -- vortex trace, in which a difference in velocities appears and forces of viscosity have a known effect on the motion, and finally, III -- the boundary layer, where as the result of a large difference in velocities at adjoining points forces of viscosity and partially forces of inertia play the primary role. There is no clear boundary between these zones.

At relatively low velocities the air or fluid in the boundary layer moves in the form of unmixed streams. This layer is called laminar. Here whirling of the stream is caused only by rotation of the liquid particles around axes perpendicular to the plane of the flow, but particles do not pass from one layer to another. An increase in velocity will lead to orderless whirling of the layer -- it changes to turbulent. The layer can also be mixed, i.e. have turbulent and laminar parts (Fig. 12, b). Under otherwise equal conditions, the kinetic energy of a turbulent flow is higher than that for a laminar one. /29

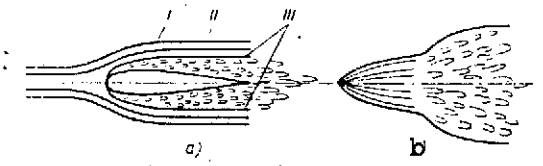

Fig. 12. Flow of a stream past a body and a mixed stream.

In the laminar layer, velocity changes according to the law of a quadratic parabola (Fig. 13), and in a turbulent one -- the velocity gradient is higher. A coefficient of resistance C_x of the body is also greater since the forces of friction are significantly higher. This is true for well streamlined bodies, where boundary layer separation does not occur. In a body with poor streamlining, the coefficient of friction also increases with an increase in turbulence, but this is caused with an expansion of the zone of stall behind the body, which will lead to a large difference in pressure before and behind the body. In well streamlined bodies, an increase in the length of the turbulent boundary layer section takes place due to the laminary one, and therefore the forces of friction and resistance to motion of the body increase.

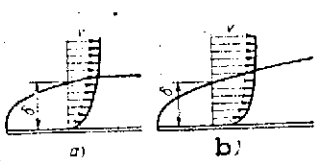

Fig. 13. Distribution of velocities in the boundary layer.

Key:
a. In a laminar layer
b. In a turbulent layer
c. Thickness of boundary layer

§ 2. Laws and Criteria of Similarity

If aerodynamic phenomena are such that their identical physical values for any point in space are located in a single relationship, they are called similar [62]. The geometric similarity of a model of nature is expressed in the proportionality of their linear dimensions l_m and l_n and in equality of angles. The relationship $k_l = l_m/l_n$ is called the linear scale of the model. The force scale k_R, the time scale k_t and others are used similarly to the linear scale k_l. For similar phenomema, the scales of identical values in similar points must be identical. If part of the scales does not fit this condition, the phenomena are partially similar.

The aerodynamic characteristics of a body are defined by a number of aerodynamic coefficients, depending on its linear and angular parameters, and also on a number of extracted numbers called criteria of similarity. Criteria of similarity consider one of the factors determining the rate of streamlining. Therefore in the modeling of similar phenomena, it is necessary also to observ equality of these criteria. In wind technology, the primary number which is used is the Reynolds number Re, which rates the viscosity of liquid, and in some cases, in the study of superfast windmills, the Mach number Ma, which reflects the effects of the resilience of the liquid is also used

$$\mathrm{Re} = \frac{vl}{\nu}; \tag{40}$$

$$Ma = \frac{v}{a}, \tag{41}$$

where v is the velocity of the stream relative to the body in m/sec; l is the linear dimension of the body (the object investigated); ν is the coefficient of kinematic viscosity; a is the speed of sound. /30

While for a model and a full-scale body the values of ν are identical, to observe similarity it is necessary that $v_m l_m = v_n l_n$. The product vl is called the characteristic of experience and is sometimes used instead of the Re number.

Therefore, during testing of a wind engine and its model in a wind tunnel it is necessary that they be geometrically similar, have identical Re numbers and identical orientation in space with respect to the flow.

Experiments have shown that in streamlining of a body the laminar flow is maintained only up to a certain Re number, and after that it transforms into turbulent flow. The Re number at which this transfer occurs is called the critical one. For plates it is

$$\mathrm{Re}_{cr} = \frac{v_0 l_{cr}}{\nu} = 4.85 \cdot 10^5,$$

therefore

$$l_{cr} = 4.85 \cdot 10^5 \frac{\nu}{v_0}.$$

For air under normal conditions

$$l_{cr} = 4.85 \cdot 10^5 \frac{1.45 \cdot 10^{-5}}{v_0} = \frac{7.05}{v_0}.$$

The Re_{cr} number depends on the initial turbulence of the flow, the greater the turbulence the lower is Re_{cr}.

§ 3. Aerodynamic Coefficients. Drag and Lifting Force of a Wing

Aerodynamic coefficients are extracted numbers whose use allows corresponding forces and moments acting on a full-scale body (object) to be approached from the forces and moments acting on the model.

In determining aerodynamic coefficients, the earth coordinate system (Fig. 14) in which oy_0 is directed vertically upward and ox_0 and oz_0 are arranged in the horizontal plane and are mutually perpendicular is usually used. Designating the velocity of the flow as the vector, and the broken line as the direction of the chord of the wing, we call angle α between them the true angle of attack, which will change upon rotation of the wing relative to axis oz.

Usually, the total aerodynamic force R acting on the streamlined flow of the body is spread out along the coordinate axes (Fig. 15), and, introducing the coefficients: C_y -- lift force, /31 C_x -- frontal drag, C_z -- lateral force, C_R -- total aerodynamic force, we get frontal drag as

$$X = C_x Sq = C_x S \frac{\rho v^2}{2} \mathbf{n} \tag{42}$$

lifting force as

$$Y = C_y Sq = C_y S \frac{\rho v^2}{2} \mathbf{n}, \tag{43}$$

lateral force as

$$Z = C_z Sq = C_z S \frac{\rho v^2}{2} \mathbf{n}, \tag{44}$$

and total force as

$$R = \sqrt{X^2 + Y^2 + Z^2}\, \mathbf{n} \tag{45}$$

where S is the wing area in m^2; $q = \rho v^2/2$ is the velocity pressure in n/m^2.

Then the coefficient of total aerodynamic force is

$$C_R = \sqrt{C_y^2 + C_x^2 + C_z^2}. \tag{46}$$

Similarly, upon the expansion of the moments, we get the roll moment

$$M_x = C_{t_x} SqL = C_{t_x} SL \frac{\rho v^2}{2} \mathbf{n} \cdot \mathbf{m}, \tag{47}$$

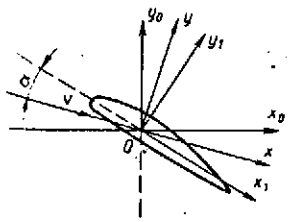

Fig. 14. Earth system of coordinates (axis z is perpendicular to the plane of the drawing).

the yaw moment M_y and the pitch moment M_Z.

Designating the arbitrary length of the body as L and C_t as the coefficient of the total aerodynamic moment, we write the basic expression for moments as

$$M = C_t S L \frac{\rho v^2}{2}. \tag{48}$$

Aerodynamic coefficients are obtained by testing models in wind tunnels. /32

On the basis of expression (42) the coefficient C_X will be

$$C_x = \frac{X_M}{S_M \frac{\rho_M v_M^2}{2}}. \tag{49}$$

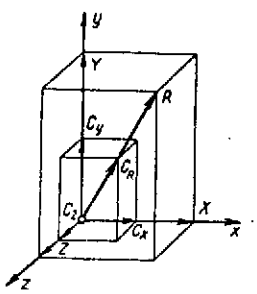

Fig. 15. Distribution of force along the axes of the coordinates.

In determining C_y and C_Z, the corresponding Y_m and Z_m are placed in the numerator. The index "m" indicates that the values relate to the model.

In order to find the force acting on the body during its flow-past by the stream, we use the expression

$$P_x = C_x F \frac{\rho v^2}{2}, \tag{50}$$

where F is the maximum area of the cross section of the body, directed perpendicular to the flow (the so-called middle section).

When the angle of attack α is changed, coefficients C_X, C_y and C_Z change.

Curves characterizing the aerodynamic properties of the wing are called aerodynamic characteristics, and those showing $C_y = f(C_X)$ at different α are called the Lilienthal polars (Fig. 16, curve 1). A line projected from the origin of the coordinates O to any point M of the polar (with identical scales for C_y and C_X), gives the value C_R, whose vector with the axis of the abscissa composes angle θ. The tangent of this angle

$$\tan \theta = \frac{C_y}{C_x} = K \qquad (51)$$

received the name lift-to-drag ratio of the wing.

The reverse relationship

$$\frac{C_x}{C_y} = \frac{1}{K} = \mu \qquad (52)$$

is called the inverse lift-to-drag ratio. In the best wing, the value of K = 22-25. /33

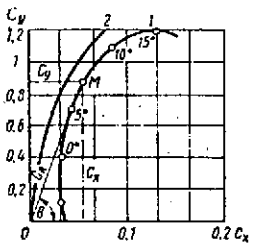

Fig. 16. Lilienthal polar and parabola of induced drag.

The total aerodynamic force intersects the axis ox connected with the wing at the center of pressure (Fig. 17). The position of the center of pressure on the wing is determined through the coefficient of the center of pressure C_p, which is usually given in percents of the chord b:

$$C_p = \frac{x_1}{b} 100\%. \qquad (53)$$

At low angles of attack ($\alpha \leq 20°$), we can consider

$$C_p \approx \frac{C_m}{C_y}. \qquad (54)$$

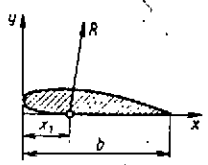

Fig. 17. Center of pressure.

The relationship of the length of the wing l_w to the chord (width) d is called the relative span of the wing λ or elongation

$$\lambda = \frac{l_w}{b}. \qquad (55)$$

For a wing with an infinite span, i.e. with $\lambda = \infty$, the resistance caused by friction of air on the surface of the wing and separation of the stream, i.e. profile drag, depends on the shape of the wing profile and the quality of its surface. Besides profile drag, during flow-past of a wing having a terminal span, additional forces of drag caused by whirling streams running off the wing arise. Therefore, with $\lambda \neq \infty$, a wing can represent a system of vortices, called connecting ones, which are attracted by the stream and form a vortex shroud behind the wing (Fig. 18, a).

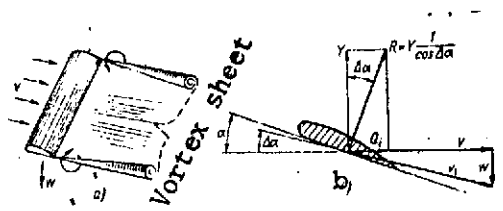

Fig. 18. Vortices running off the wing and the forces of induced drag.

The whirling lines give a vertical velocity w, which is called the downwash velocity, acting on the stream and deflecting it so that angle α is reduced by the angle of downwash Δα (Fig. 18, b). Since the direction of the lifting force is changed simultaneously, an additional force of resistance Q_1, called the induced resistance, arises. On the basis of the theory of induction, it has been established that /34

$$w = \frac{C_y}{\pi\lambda} v \text{ m/sec;} \tag{56}$$

$$\Delta\alpha = \frac{C_y}{\pi\lambda} \text{ rad;} \tag{57}$$

$$Q_i = \frac{C_y^2}{\pi\lambda} S \frac{\rho v^2}{2} \text{ n,} \tag{58}$$

where $C_y^2/\pi\lambda = C_i$ is the coefficient of induced drag.

We call the curve $C_i = f(C_x,\alpha)$ the parabola of induced drag (see Fig. 16, curve 2). The total aerodynamic characteristics of profiles of wings having a terminal span are presented in appropriate aviation atlases.

In streamlining a wing with an airstream, a zone of reduced pressure is created above it and a zone of increased pressure is created beneath it (Fig. 19, a), which causes the appearance of the lifting force. According to the Bernoulli theorem, we assume that the velocity of the stream above the wing is higher and that below the wing is lower than the velocity of particles away from the wing. Consequently, circulation velocity (Fig. 19, b) around the wing and vortex line must exist. N. Ye. Zhukovskiy proposed in calculations substituting the wing with a sheaf of vortices, which he called attachments, and introduced a formula for the lifting force of a wing having an infinite span

$$Y = \rho F v_0 l_W \tag{59}$$

where Y is the force perpendicular to the velocity v_0 of the undisturbed stream and acting on wing section of length l_W.

The appearance of this force is caused not only by the difference in pressures but also by the impulse of the force, i.e. the appearance in the fluid of a quantity of pressure. In an ideal liquid, the force of frontal drag X of the wing is equal to zero.

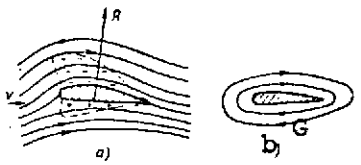

Fig. 19. Lift force of a wing (a) and circulation velocity around its profile (b).

The theorem of N. Ye. Zhukovskiy can be stated thus: the lift force on the wing with $\lambda = \infty$ is equal to the product of the density of the liquid, the circulation of the velocity, the velocity of the stream in infinity and the length of the isolated wing section. With this, the direction of the force is obtained by rotating the velocity vector by an angle of 90° towards the side opposite the direction of the circulation of velocity.

With $\lambda \neq \infty$, the part of the liquid streamlining the wing which /35 is included between the vortices picks up in the vertical direction an increase in the quantity of pressure, which is measured by the value $\rho G l_w$.

For practical calculations, the equation of the connection between the theoretic values of circulation and the experimentally determined coefficient C_y is used:

$$G = \frac{C_y}{2} b v_0. \qquad (60)$$

From examination of the above, it is clear that wings with different elongations have different Lilienthal polars. The characteristics of the profiles, obtained in the wind tunnel, are given for $\lambda = \infty$. When using them for designing windmills with the graphical method, the polars for a wing of infinite span are recomputer [35].

Chapter 3. Theory of the Wind Engine

The majority of wind engines are equipped with a windmill, located in the vertical plane, and having blades which have an aerodynamic profile. The axis of rotation of the windmill is, as a rule, horizontal (it is sometimes located at an angle of 5-10° from the horizon) and parallel to the direction of the stream. Such engines are called vaned ones; they will also be examined further on. We will introduce the following concepts.

Coefficient of utilization of wind energy ξ -- the relationship of a second's work, accomplished by the windmill, to the energy of the flow passing in a second through a section equal to the surface swept by the windmill.

Ideal windmill -- windmill with an axis of rotation parallel to the velocity of the wind, having an infinitely large number of verynarrow blades, whose profile drag is equal to zero and the

circulation of velocity along the blade is constant, rotating with an infinitely high angular velocity and, finally, being distinguished by the fact that the loss in airflow velocity in the plane of the windmill's rotation is constant along the entire swept surface.

Actual windmill -- windmill having a terminal number of blades and angular velocity ($\omega_{wm} \neq \infty$) and working with losses, chief among which are: tip, profile, from twisting of the stream behind the wheel and from partial utilization of the swept surface due to the presence of space near the hub which is not occupied by blades.

§ 1. Theory of the Ideal Windmill

/36

The theory of the ideal windmill was first developed by V. P. Vetchinkin in 1914 and after 6 years N. Ye. Zhukovskiy gave the solution for the coefficient ξ and founded a number of important propositions of the ideal windmill.

The classical theory of N. Ye. Zhukovskiy establishes that for an ideal windmill $\xi_{max} = 0.593$. According to the theory of G. Kh. Sabinin, which is more convenient for practice, $\xi_{max} = 0.687$. This discrepancy was caused by the fact that in determining the axial force of pressure of the flow on thewindmill, G. Kh. Sabinin determined the impulse of the forces not at the place where the vortex solenoid was formed, but in the place where it adopted a cylindrical form and has an area larger than the swept surface.

Let us assume that a stream flows toward the windmill with a velocity v in section I-I (Fig. 20, a). Then in section II-II, the velocity $v_2 = v - v_{II}$, and in section III-III, $v_3 = v - v_{III}$. Due to the pressure head in front of the rotating windmill, velocity v of the stream will fall, and then after transversing the section of the windmill will almost not change (Fig. 20, a, curve 1), pressure, on the contrary, will increase in front of the windmill and will sharply drop upon transversing its section, and, according to its distance from the windmill, will asymptotically approach zero, i.e. normal pressure will be reestablished (Fig. 20, a, curve 2).

The difference in reserves of kinetic energy of the flow in front of the windmill $mv^2/2$ and behind it $m(v - v_{II})^2/2$ is transferred to the windmill and if there is no loss, the useful work can be expressed as

$$T_1 = \frac{mv^2}{2} - \frac{m(v-v_{III})^2}{2} = mv_{III}\left(v - \frac{v_{III}}{2}\right). \qquad (61)$$

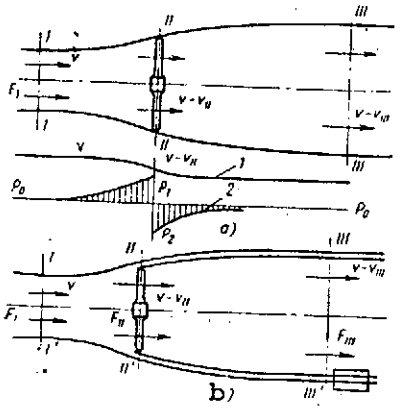

Fig. 20. Character of an air flow penetration through a windmill: a -- according to the classical theory; b -- according to the theory of G. Kh. Sabinin.

On the other hand, the energy absorbed by the windmill is equal to the product of the force of pressure of the wind P and the velocity in the plane of rotation:

$$T_1 = P(v - v_{II}). \qquad (62)$$

Equating expressions (61) and (62), /37 after simple transformations, we obtain

$$v_{III} = 2v_{II}.$$

In this way, the total loss in velocity behind the windmill is twice as large as the plane of its rotation, which in turn comprises one-third of the velocity of the wind.

We write the relationship of a second's work of the ideal windmill to the energy of the flow with the area of the section as equal to the area of the swept surface F_{II}, and obtain the value

$$\xi_i = \frac{P(v + v_{II})}{F_{II}\frac{\rho v^3}{2}} = 2\frac{P}{F_{II}\rho v^2}\cdot\frac{v - v_{II}}{v}. \qquad (63)$$

The expression $\frac{2P}{F_{II}\rho v^2}$ is designated as V and called the coefficient of frontal pressure; we also introduce the coefficient of braking $e = v_{II}/v$. Making the necessary substitution and differentiating the final expressions for the purpose of finding e corresponding to $\xi_{i\ max}$, we obtain

$$B = 4e(1 - e); \qquad (64)$$
$$\xi_i = 4e(1 - e)^2. \qquad (65)$$

With e = 0.333, $\xi_{i\ max}$ = 0.593, and B = 0.888.

According to the theory of G. Kh. Sabinin [29], the diagram of passage of the flow through the windmill differs from the classical. The surface limiting the flow has the shape shown in Fig. 20, b; its sections along the entire length are circular. The part of the flow limited by this surface is called the working

flow. Along the surface of the segment in section II-II'-III-III' there is a sort of vortex solenoid, whose streams flow parallel to each other and whose pressures at all points of the flow which are sufficiently removed from the windmill, are constant. An increase in the quantity of motion created by the windmill will be equal to the quantity of motion formed by the solenoid.

While in section I-I' (Fig. 20, b) the stream has a velocity v and a cross sectional area F_I, in section II-II', the area increases to F_{II} (the so-called swept surface), and in section III-III', relatively removed from the windmill, the velocity of the stream will be $v - v_{III}$ inside the solenoid and v outside it, and the section area is F_{III}. Velocities v_{II} and v_{III} are caused by the vortex solenoid on the windmill and at a sufficient distance from it respectively.

Having determined the circulation of the velocity inside the vortex solenoid and the impulse of force on the windmill over time dt, we obtain the expression for P in its final form

$$-P = [\rho F_{III}(v - v_{III})]\, v_{III} + \left(\frac{\rho F_{III}\frac{v_{III}^2}{2}}{v_{III}}\right) v_{III} = (m_1 + m_2)\, v_{III}, \qquad (66)$$

where $m_1 = \rho F_{III}(v - v_{III})$ is the mass of air passing through the /38 swept area in a unit of time; $m_2 = \frac{\rho F_{II}\frac{v_{III}}{2}}{v_{III}}$ is the <u>attached mass of air</u>.

The sum $m_1 + m_2$ is called the <u>entrapped mass</u> and for set conditions remains constant in value.

Since according to the theory of G. Kh. Sabinin the swept area F_{II} is the arithmatic mean of the areas of the working stream in front of F_I and behind F_{III} of the windmill

$$F_{II} = \frac{F_I + F_{III}}{2},$$

which by comparison with the classical theory changes the values ξ_{max} and B. With this, the values $\xi_{i\ max} = 0.687$ and $B_{max} = 1.172$ corresponds to the coefficient of braking e = 0.414. The velocity of the stream behind the windmill decreases by the amount

$$v_{III} = \frac{2v_{II}}{1 + \frac{v_{II}}{v}}, \qquad (67)$$

the axial (frontal) pressure on the windmill is

$$P = (m_1 + m_2)\, v_{III} = F_{III} \rho v v_{III}. \tag{68}$$

§ Theory of an Actual Windmill

The theory of the actual windmill, developed by G. Kh. Sabinin, emanates from the proposition that the difference in pressures of the airflow on both sides of the windmill, acting on the area of a circle obtained by the intersection of the airstream and the swept surface, is absorbed by the elementary blades (Fig. 21). On the basis of this proposition and examination of a plan of velocities of the stream rushing toward the element of the blade (Fig. 22), two equations of interrelationship were developed.

The first equation relates the width of the blade and the coefficient of lift force with the formation of the stream, which is characterized by value e.

$$\frac{i_n b C_y}{8\pi r} = \frac{e}{(1+e)(1-e)^2} \cdot \frac{1}{(Z_u + \mu)\sqrt{1+Z_u^2}}, \tag{69}$$

where i_n is the number of blades in the mill; b is the blade width along the chord in m; r is the flow radius at which the wing section being examined is located, in m.

The second equation provides the possibility for determining the number of relative modules $Z_u = \cot \beta = (\omega_{wm} r + u_1)/(v - v_1)$ with a known speed of the windmill $Z = (\omega_{wm} R)/v$ and braking e. /39

$$Z_u = \frac{1}{2}\left[\frac{Z}{1-e} - \mu\left(1 + \frac{e}{1-e^2}\right)\right] \pm \pm \sqrt{\frac{1}{4}\left[\frac{Z}{1-e} - \mu\left(1 + \frac{e}{1-e^2}\right)\right]^2 + \frac{e}{1-e^2} + \mu\frac{Z}{1-e}}, \tag{70}$$

where R is the radius of the windmill in m; ω_{wm} is the angular velocity in rad/sec.

From Fig. 22 is is also evident that the optimal velocity of the flow moving toward the blade element will be

$$w = \sqrt{v_1^2 + (-\omega_{wm} r - u_1)^2} \quad \text{m/sec} \tag{71}$$

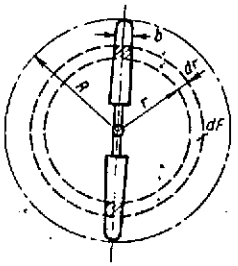

Fig. 21. On defining the concept of the elementary blades.

where u_1 is the reaction velocity from the torsion moment, developed by the blades and corrected against the moment; v_1 is the velocity of the stream in the plane of the windmill.

Finally, angle ϕ of blade inclination and radius r will be

$$\phi = \operatorname{arccot} Z_u - \alpha. \qquad (72)$$

The major characteristics of a windmill include the moment and power developed by it. If the moment of elementary blades of length dr is integrated within limits of from r_0 up to R, we obtain the total moment developed by the windmill

$$M = \int_{r_0}^{R} dM = \int_{r_0}^{R} 4\pi r^2 \rho \frac{e}{1+e} v^2 \frac{1-\mu Z_u}{Z_u+\mu} dr. \qquad (73)$$

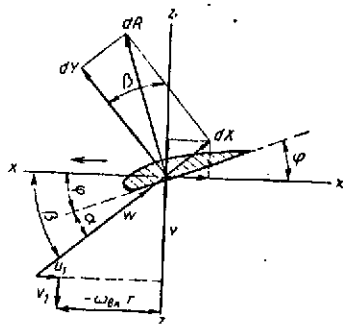

Fig. 22. Plan of airstream velocities and forces acting on the blade.

It is convenient to express the moment in the form of an extracted value /40

$$\overline{M} = \frac{M}{\pi R^3 \rho \frac{v^2}{2}} = \int_{\bar{r}_0}^{\bar{R}} 8 \frac{e}{1+e} \cdot \frac{1-\mu Z_u}{Z_u+\mu} \bar{r}^3 d\bar{r}, \qquad (74)$$

where $\bar{r} = r/R$ is the relative radius.

Disregarding torsion of the stream and considering e constant along the radius, we obtain

$$\overline{M} = \frac{4e}{Z} \cdot \frac{1-e}{1+e} \left[\left(1 - \frac{r_0^2}{R^2}\right) - 2\mu \left(\frac{Z_u}{3} + \frac{1 - \frac{r_0}{R}}{Z_u} - \frac{1 - \frac{r_0^2}{R^2}}{2} \right) \right]. \qquad (75)$$

Since $M = \overline{M}\pi R^3 \frac{\rho v^2}{2}$, a $\omega_{wm} = Z_v/R$, the power developed by the windmill will be

$$N_{wm} = M\omega_{wm} = \overline{M}\pi R^3 \frac{\rho v^2}{2} \omega_{wm} = \overline{M}\pi R^2 \rho \frac{v^3}{2} Z \quad \text{n·m/sec} \qquad (76)$$

We note that in the best high-speed windmills, ξ_{max} = 0.45-0.48 and in low-speed ones, ξ_{max} = 0.35-0.38.

§ 3. Aerodynamic Characteristics of the Wind Engine

An aerodynamic characteristic shows the dependency of the extracted (relative) moment $\bar{M}$ and the coefficient of utilization of wind energy ξ on the number of modules. Its form depends on the number, width and aerodynamic profile of the blades and the angle of inclination. The curve $B = f(Z)$ is also given frequently.

The major points of the curve defining the major parameters of the characteristics (Fig. 23), will be: $\bar{M}_n$ is the nominal relative rotating moment, which the windmill develops with the nominal number of modules Z_n; $Z_n = (\omega_{wm} \cdot nR)/v$ is the turning speed (number of modules) at which $\xi = \xi_{max}$; $\bar{M}_0$ is the initial relative moment developed by the wheel during starting, i.e. when $Z = 0$; Z_0 is the synchronized speed, at which $\bar{M} = 0$; $\bar{M}_{max}$ is the maximum moment developed by the windmill. The relationship $\bar{M}_{max}/\bar{M}_n$ is called the overload of the windmill.

From examining the characteristics of windmills having different turning speeds (Fig. 24) and also the results of theoretic and experimental investigations, the following conclusions can be drawn. /41

1. The greater the number of blades, their width and angle of inclination, the lower the turning speed of the windmill and the value of Z_0, and the higher the value of $\bar{M}_0$ and the curve $\bar{M}(Z)$ has a sharply dropping form.

2. In high-speed mills, the value of $\bar{M}_0$ is several times smaller than $\bar{M}_n$, and the synchronized speed is 2-2.5 times higher than normal.

3. The power of the windmill under otherwise equal conditions depends little on the number of blades and the coefficient of fill $k_f = S/F$, which is the relationship of the blade area to the swept surface. The form and profile of the blades, their position in the airflow and the diameter of the windmill have the major effect on power.

4. The reduction in the relative initial moment occurs more intensively than an increase in speed. Thus, with an increase in Z_n by twice, $\bar{M}_0$ decreases by 6-7 times. At the same time, pickup, defined by the relationship $\bar{M}_0/M_n$, in a six-bladed wheel is 3.3 times greater than in a two-bladed one.

5. For the constant aerodynamic diagram, frequency of rotation of the windmill is directly proportional to its speed and wind velocity and inversely proportional to diameter.

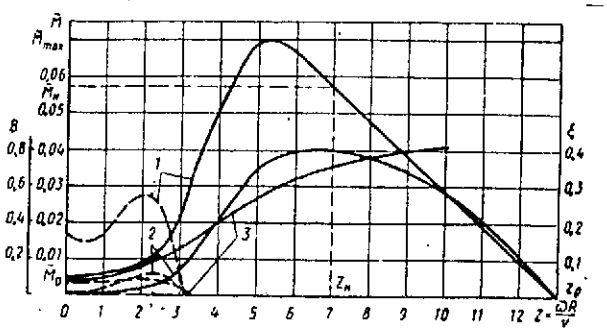

Fig. 23. Aerodynamic characteristics of two-bladed windmill ($Z_n = 7$): 1. $\bar{M}(Z)$; 2. $\xi(Z)$; 3. $B(Z)$ (solid lines -- with a calculated angle $\phi_p = 8°$; broken lines -- with a starting angle $\phi_n = 31°$).

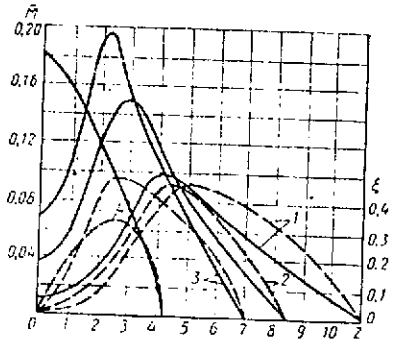

Fig. 24. Characteristics of windmills of various speeds: 1. two-bladed; 2. three-bladed; 3. six-bladed; 4. 18-bladed [solid lines $\bar{M}(Z)$; broken lines $\xi(Z)$].

A change in some dimensionless and relative aerodynamic parameters of windmills, represented in portions of the values of the same parameters of a two-blade with $Z_n = 7$, indicated by index $i = 2$ is shown in Fig. 25. The blades of the wheel, with the exception of the 18-bladed one have a "Espero" profile. It is not difficult to be convinced that due to the large value for $\bar{M}_0$, it is advantageous to use slow-running mills. With the same graident of increase, the relationship $\bar{M}_0/\bar{M}_{0i} = 2$ is greater than the reduction in Z_n. However, with a reduction in Z_n from 5 to 2.7 (by 1.85 times), $\bar{M}_n$ increases by only twice. Therefore, from the point of view of the value of the nominal moment, the use of slow-running mills does not give any advantage [51]. In a number of cases (for instance, during work with generators and compressors) under comparable conditions, the outputs of high-speed and low-speed wind engines are approximately identical. /42

When selecting and computing layouts for windmills, it is necessary to consider the aerodynamic characteristics of windmills and the loading characteristics of the working machines. For instance, for driving a low-speed pump having a high initial moment of resistance, it is better to use a multibladed windmill; this allows a reduction in the value of the minimum working velocities of the wind, and a reduction in the gear ratio of the reduction gear and in mechanical losses. True, by comparison with the high-speed windmill, the value of ξ_{max} is reduced. For combination with a generator having a high frequency of rotation, a low initial moment of resistance and a smooth increase in resistance, it is more effective to use a high-speed windmill. /43

During recent years, due to the use of blades with highly improved aerodynamic profiles and different automatic systems,

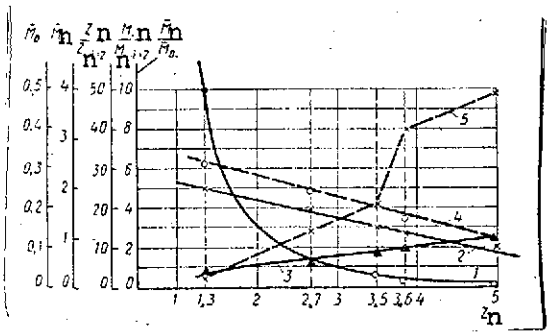

Fig. 25. Change in some dimensionless and aerodynamic characteristics of windmills: 1. $\bar{M}_0$; 2. $\bar{M}_n$; 3. $Z_n/(Z_{n/i} = 2)$; 4. $M_n/(M_{n/i} = 2)$; 5 $M_n/\bar{M}_0$.

an increase in the effectiveness of utilization of high-speed windmills have been successfully achieved, which was in part expressed in an improvement in their starting characteristics, an increase in power, a reduction in unevennes of rotation and so forth. All this has increased economy of wind machines.

PART 2. LAYOUTS AND INSTALLATION OF WIND MACHINES /44

Chapter 4. Optimization of Wind Machine Parameters

§ 1. Tasks of Parameter Optimization

The effectiveness of a wind energy-producing machine, along with its aerodynamic characteristics and design features, is affected by the aggregate regional conditions for which it is designed. These include wind rate, hydrogeological conditions, characteristics and requirements of the energy consumer and others. These conditions are distinguished by their great variation and therefore a system consisting of machines of the appropriate types and dimensions, with parameters which are optimal for the set zones and regions is necessary.

Due to conditions of production and operation, the number of types of machines must be limited within rational limits, since otherwise the series production ofeach type would be decreased, which will lead to an increase in initial costs and make more difficult the provision of spare parts due to the increase in their nomenclatures. On the other hand, a limited number of machine types (by comparison with the calculated number), whose parameters deviate from the optimal ones for the set conditions will cause some increase (by comparison with the optimum variation) in costs of providing the consumer with power.

The optimum number of standard types and their scaling into rows is established by comparing these two opposing tendencies with extraction of statistical data on the characteristics of the consumers. For instance, for windmill pumping machines, these data are the density or distribution of depths and yield of sources, graphs of water consumption, its demand, etc.

The problem represents significant difficulty even while basing just the scaling of diameter: the variations of optimal values of D_{wm} can be so multitudinous that the necessity to group machines according to types with consideration for conditions of those regions in which they will be used in the first order arises. And these conditions are extremely varied. The zones of the most density in calculated diameters can be examined beginning with the necessity for rounding off the value of D_{wm} to 0.5 m, as required by GOST [All-Union State Standard] 2656-55. The most difficulty in solving this problem, however, arises as the result of the absence of sufficient statistical data for es- /45
tablishing initial dependencies of cost as a function of production and usage conditions.

The summary adjusted expenses $\bar{E}$ for development of energy depends on the probability of using the wind machines under various wind, hydrogeological and economic conditions for various consumers and also on expenses for production of equipment. If we

have the statistical order of distribution p(L) of the generalized characteristic L, which determines conditions of utilization of the wind machine, one of the nominal parameters of which is S, then within the range L_{min}-L_{max}, it is possible to imagine n standard dimensions of machines of a set type, whose parameters will be equal to S_1, S_2..., S_n respectively.

For each S_i, there is a solution which gives the minimum summary expenses for the development of energy:

$$\bar{E} = f_i(L, S_i) \to \min.$$

If the dependency F(n) characterizing expenses on the manufacture of n machines of a certain type is known, the optimal number n of machines of the order can be established, finding the minimum summary function of the type

$$F(n) = \sum_{i=1}^{n} \bar{E}_i + F(n) \to \min. \qquad (77)$$

The graphical interpretation of the problem (Fig. 26) shows in particular that with unification of construction, the optimal number of types increases and, consequently, high possibilities arise for reduction of expenses per unit of energy developed or per cubic meter of water lifted.

Any technical optimization almost always follows the final goal -- the economic optimization, and therefore in the overwhelming majority of cases the main criterion in selecting a technical solution is its economic expediency. We will consider to be optimum a machine which under set conditions provides the minimum value for adjusted (calculated) expenses with indentical satisfaction of the energy consumer.

$$\bar{E} = U + \rho_n K \text{ rub}, \qquad (78)$$

where U is the annual outlay with normal operation of the wind installation, including deductions for renovation, in rubles; K is the one-time capital investiment in rubles; ρ_n is the coefficient of adjustment (normative coefficient of effectiveness), which in energetics is taken as equal to 0.15. /46

This approach to the selection of an optimal wind machine allows attainment of the real most economic solution. It can be effectively realized with utilization of the methods of mathematical modelling, which allow combination in a single system of the technical characteristics and economic indices which most

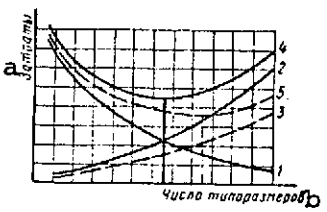

Fig. 26. Toward determining the optimum number of wind machine types: 1. Expenses in the sphere of utilization; 2. Same, in the sphere of production; 3. Same, with unification; 4. Summary expenses; 5. Same, with unification.

Key: a. Expenses; b. Number of types

fully reveal the basic dependencies characterizing the machine. In combination with the use of electronic computer equipment, this provides (by means of a directed multivariational excess) location of the real optimal machine of a large number of possible ones. The quantitative data obtained must be the basis for accepting the solution before designing, for tuning the adjusting system, selecting the gear ratios between the windmill and working machine assembled with it, for establishment of base wind velocities: minimum v_0 and maximum working velocity $v_{\text{work max}}$, as well as the maximum for strength calculations $v_{c\ \max}$.

During creation of wind engines and their assembly with working machines, the required attention is not always paid to variational energy balance and technical-economic calculations. The use of methods of mathematical modelling, providing the possibility for developing the basic regularities and interdependencies of parameters which determine the design, technical and economic indices of the machines, allows forecasts to be made as to the direction in which they should be changed, in order to obtain the best results and steer away from passive statistical registration of results already on hand from designs previously accomplished and move to a more goal-oriented analysis for the purpose of improving the machine, and increasing its reliability and effectiveness.

§ 2. Mathematical Modelling of Wind Machines

Differing from a physical model, which is similar to the initial (investigated) machine, the mathematical model describes the machine using a system of equations which relatively fully reflects the relationship of the basic parameters and allows judgment to be made of their effect on the major indices of the work of the machine. However, not every mathematical model will yield the desired results from the point of view of economic effectiveness of the machine. In particular, if the problem of determining under set conditions the maximum output in cases where the entire output can be used is set up, the solution is relatively simple and boils down to finding the maximum of the function

$$v^3 = \int_{x=v_0}^{x=v_{\text{n.reg}}} f(x)x^3\,dx + \int_{x=v_{\text{n.reg}}}^{x=v_{\text{work max}}} f(x)c_i^3\,dx \to \max, \qquad (79)$$

where f(x) is the density of distribution of wind velocities; /47
f(x) dx is the probability of the fact that the value of v is within the limits of from x to (x + dx); $c_1 = v_{cal}$ is the calculated velocity of the wind with which the established power of the wind engine is determined, in m/sec.[9].

With this, we can vary only values v_0, $v_{n.reg}$, and $v_{work\ max}$ for the purpose of finding the best variation for which output $A = A_{max}$ without considering criteria of effectiveness. The deficiency in this model consists of the fact that obtaining the maximum output does not provide the maximum economy of the machine.

We will look at a wind energy-producing installation as a complex system, consisting of a machine, a reserving or duplicating device, automatic control mechanisms and the consumer. For pumping installations, a water source is also included in the complex.

The single technical and economic relationships between the engine, load, storage cell and so forth characterize the installation. These complexes can be called static, since they are usually studied as applicable to one season or year. It is competent to examine inside this complex at the beginning optimization of parameters of separate elements, in the first place of the wind machine, considering not only internal, but also external relationships. With this, the installation can be included in the class of combined systems, in which some parts interact in a clear, definite manner (determinate systems), and others are probable. This is understandable since, for instance, the rates of alternation in wind intensity, and consequently, the power developed by the machine are random and forecasting them for an extended period is difficult. They are, however, to a certain degree subjected to statistical conformities.

The overall problem of modelling and localizing the optimal parameters is divided into partial problems with a smaller number of variables. Optimization of the wind machine is an important stage, necessary for the achievement of the high economy of the installation. With this, the machine is a determinate system, whose final effect is determined by expenses and the summary output over the calculated period. We will examine a mathematical model for an example of a pumping wind machine. All the basic reasoning also remains justified for machines of other types.

The model must as fully as possible reflect the relationship between the parameters, their effect on the work of the machine and the technical and economic indices, and, chiefly, provide the possibility for locating their values at which $\bar{E} = \bar{E}_{min}$. For the purpose of simplifying the model and subsequent solutions, we will omit, as is usually done, the second degree dependencies, which have little effect on the precision of the results obtained.

One of the features of the method of calculating expenses is /48
that the results of comparing several variations are not changed if those components which are identical for the variations being examined are excluded from the expenses. For this reason, the realtionship between the parameters and expenses can be seen by considering only the expenses depending on the parameter investigated. Therefore, making a number of assumptions, the value of specific calculated expenditures $\bar{E}$ can be considered equivalent to the specific cost of the machine. This simplifies the program and condenses the scope of computations, providing at the same time sufficient precision and reliability in the calculations for practice.

Indeed, the specific adjusted (calculated) expenditures $\bar{E}$, related to one kW·h of developed energy or to 1 m^3 of water lifted over the calculated period by the machine, according to formula (78) are equal to

$$\bar{E} = \rho_n \frac{K}{Q_\Sigma} + \frac{U}{Q_\Sigma} \text{ rub/m}^3 \text{ or rub/kW·h}, \qquad (80)$$

or, equalizing the summary output Q_Σ to water consumption B_Σ,

$$\bar{E} = \rho_n \frac{K}{B_\Sigma} + \frac{U}{B_\Sigma} \text{ rub/m}^3. \qquad (81)$$

In cases where the wind-electric machine works parallel with a circuit, its entire output $A_{0\ max}$, determined by the wind rate, can be effectively and totally utilized. Therefore, the value of $A_{0\ max}$ should be substituted in the denominator of equation (80).

Formulas (80) and (81) are justified for cases in which the capital investiments K are one-time (i.e. the construction period of the object is not more than one year) and the annual production outlays are constant, which usually takes place during construction of wind installations. In this manner, after starting the installation into operation, further capital investments are not produced.

Annual outlays U are made up of amortized deductions for renovation U_r and major repair U_{mr} of the machine, expenses for acquiring polishing and lubricating materials U_{pl} and expenses for operation U_o. For wind machines in operation at the present time, the overall norm of amortized deductions is 10.2%, of which 7.2% is for renovation. Expenditures for lubricating and polishing materials are low and make up part of a percent of the total outlays. Finally, for wind installations of definite purpose, the time spent on servicing and the wages of the mechanic do not differ extremely. This is especially characteristic for group servicing -- a method being ever more widely utilized.

Therefore, with accuracy sufficient for practice, it is possible to consider annual expenses to be proportional to the cost of the machine C_{ma}, i.e.

$$U = k'_{pr}C_{ma}, \tag{82}$$

where k'_{pr} is the coefficient of proportionality.

Capital investments are also proportional to the cost of the machine: /49

$$K = C_{ma} + C_a + C_{tr}, \tag{83}$$

where C_a is the cost of assembly, usually equal to (0.1-0.12) C_{ma}; C_{tr} is the transport costs, depending on the weight and consequently, on the cost of the machine.

Then, with an error of no more than 1-2%,

$$K \approx k''_{pr}C_{ma}, \tag{84}$$

where k''_{pr} is the coefficient of proportionality.

Now the equation for determining $\overline{E}$ can be presented as:

$$\overline{E} = \rho_n \frac{k''_{pr}C_{ma}}{B_\Sigma} + \frac{k'_{pr}C_{ma}}{B_\Sigma} = \frac{C_{ma}}{B_\Sigma}(k'_{pr} + \rho_n k''_{pr}). \tag{85}$$

Since k'_{pr}, k''_{pr} and ρ_n are constant values for a machine having a definite purpose, then designating $(k'_{pr} + \rho_n k''_{pr}) = M$, we obtain

$$\overline{E} = M\frac{C_{ma}}{B_\Sigma}, \text{ or } \overline{E} = \frac{C_{ma}}{A_{o\ max}}. \tag{86}$$

Therefore, the machine whose cost C_{ma}, related to the total development of energy (for a pumping machine, to a set water consumption B or to the total quantity of water lifted Q_Σ) over the calculated period, is minimal, which also corresponds to the minimum $\overline{E}$ as the optimal paramters:

$$\overline{E} \equiv \overline{C}_{ma} = \frac{C_{ma}}{Q_{\Sigma}(B_{\Sigma})} \rightarrow \min, \text{ or } \frac{C_{ma}}{A_0 \max} \rightarrow \min. \qquad (87)$$

We will call the criterion $\overline{C}_{ma}$ the specific cost of the machine.

If in machines $\overline{C}_{ma}$ are identical, the secondary, auxiliary criterion of optimality can be considered to be the specific metal consumption

$$\overline{G}_{ma} = \frac{G_{ma}}{Q_{\Sigma}(B_{\Sigma})} \rightarrow \min, \text{ or } \frac{G_{ma}}{A_0 \max} \rightarrow \min. \qquad (88)$$

The task of optimization boils down to locating the minimum of the purpose function

$$\min\left[\frac{F_1(k_n, D_{wm}, h_{opzn}}{F_2\left(C \int_{v_*}^{v_{work\ max}} v_i^3 dv_i f_{v_i} \xi_{v_i}\right)}\right], \qquad (89)$$

where F_1 is the cost of the machine, determining its construction parameters and technical characteristics; F_2 is the energy consumption over the calculated period or output of the machine over the same period, depending on wind rate and productivity of the machine under set conditions; C, k_n are the coefficients depending on the type and construction of the machine. /50

The model of a pumping unit, after some transformation, can be represented by a system of 15 equations, relating its basic parameters. The solution of problems on optimization foresees the location for set conditions (region and period of utilization of the machine, energy consumption) of those values of the diameter of the windmill, the height of the support, base wind velocities, and gear ratio functions of the drive, at which $\overline{C}_{ma}$ has the minimum value. The precision of the results obtained depends on errors in initial information and those admitted during solution of the problem. In particular, it is impossible to accurately consider the actual repetition of wind velocities, and therefore, generalized distribution curves for the zones, corresponding to average velocities over many years, and, consequently, having a 50% norm of guarantee are used.

In calculating pumping installations, it is necessary to operate with an averaged norm for water consumption, although there are some fluctuations in consumption which are difficult to consider and depend on the season, weather conditions and actual

head count of livestock. The utilization of averaged values for technical indices of the machine, while they can change depending on the degree of design perfection of the windmill, drive, water lift and the quality of their assembly, and averaged data on expenditures for manufacture of components, while indices characterizing capital investments can within some limits vary depending on technology of production and size of series, also yields some error.

§ 3. Methodology of Optimizing Water-Lifting Wind Machines

The methodology for optimization of parameters is based on the comparison of the summary output of the machine, equated to water consumption, with the expenditures which must be carried with it. To locate the optimal solutions, it is most convenient to use the method of directed comparison of variations, using electronic digital computer equipment for the calculations.

The search for optimal combination of the basic parameters of the machine, corresponding to the condition $E = E_{min}$, requires solution of a system of equations in which variable values which are frequently vaguely related to each other are introduced.

For instance, the overall weight of the tower comprises up to 30% of the total weight of the machine. Reducing the weight of the tower and, consequently, the cost of the machine, is possible, /51 using a rational layout for the spatial grid and decreasing height. In the latter case, lightening of the tower is also achieved as the result of decreasing the tipping moment created by aerodynamic forces. At the same time, the opposite tendency, raising the windmill to a greater height, where the intensity of the wind is higher and its flow is more even, is natural. It is possible to presume the presence of the value for h_{op}, which provides with otherwise equal conditions the attainment of the minimum value for $\bar{C}_{ma}$.

We will look at another example. The values for the base velocities of the wind have a great effect on the annual output of the machine. It is desirable to decrease the value of v_0, but with set characteristics of loading, primarily M_0, and the selected aerodynamic layout of the windmill, this indicates an increase in D_{wm}, and consequently in its weight and loading. On the other hand, in order to increase the output, it is possible to use higher values for $v_{n.reg}$ and $v_{work\ max}$, but this will also lead to an increase in calculated loads, and as a result, to an increase in metal consumption and cost. The search for the optimal combination of v_0, $v_{n.reg}$ and $v_{work\ max}$ will also give their values at which $\bar{C}_{ma} = \bar{C}_{ma\ min}$.

The solution of problems of optimization, like the problems of extremals, in development of a new machine for set conditions

will consist of determining the values of the basic parameters, beginning with the requirements of optimality: for machines on hand, exposing for a set region the values of parameters relating to drive and adjustment of the regulator.

In solving the first problem with the methodology, utilization of an algorithm, whose bases are expressions (90)-(102) is foreseen. The energetic value of the wind defines the rate of distribution of velocities. The number of hours of activity of the wind at various velocities, necessary for calculations, is taken from tables and histograms constructed according to actual values of observed velocities, or, finally, are found from mathematical dependencies describing density of velocity distribution (see expressions (11) and (13)).

The <u>useful power</u> developed by an engine is

$$N_n = \frac{\rho F v_i^3 \xi_i \eta_{oa\,i}}{2 \cdot 75 \cdot 1{,}36} \approx 4{,}81 \cdot 10^{-4} D_{wm}^2 v_i^3 \xi_i \eta_{oa\,i} \text{ kW}. \qquad (90)$$

where $\eta_{oa\ i}$ is the overall coefficient of efficiency of the machine.

The coefficients ξ_i and $\eta_{oa\ i}$ are variable values depending on velocity and load. The characteristics of loading are usually known; consequently, it is possible to set up dependencies $\xi_i(v_i)$ and $\eta_{oa\ i}(v_i)$ for set conditions (Fig. 27). The values of ξ_i and $\eta_{oa\ i}$ for pumping machines of various types (Table 6) are presented in Table 7.

The <u>possible (theoretical) total output</u> is located from the combined solution of equations (11) or (13) with equation (90): /52

$$A_{O\max} = \sum_{v_0}^{v_i = v_{work\ max}} \frac{\rho F v_i^3 \xi_i \eta_{oa\,i}}{2 \cdot 75 \cdot 1{,}36} t_{v_i} = B_A D^2 \left(\sum_{v_0}^{v_{n.reg}} v_i^3 \xi_i \eta_{oa\,i} t_{v_i} + v_{n.reg}^3 \xi_{max} \eta_{oa\,max} \sum_{t_{v_i} = t_{v\,n.reg}}^{t_{v\,work\ max}} t_{v_i} \right) \text{ kW·h} \qquad (91)$$

where B_A is the constant coefficient.

If the service life of the machine is T_s, the output over the period of operation is

TABLE 6. TYPES OF MACHINES

Designation of machine	Type	Number of blades	ξ_{max}	$\bar{M}_0$
Low-speed, with belt water lift (prototypes VP-3M and VTL-3)	I	18	0.3	0.33
Dow-speed, with sucker-rod pump (prototype "Chayka")	II	12	0.33	0.25
Medium speed, with belt water lift (prototypes "Buran" and VPL-5)	III	6	0.38	0.07
High-speed, with pneumatic pump (prototype "Lastochka")	IV	3	0.4	0.23
High-speed, with pneumatic pump (prototype -- "Vikhr'," with starting spring)	V_{pn}	2	0.43	0.016
High-speed, with electrical centrifugal pump (prototype "Berkut" with starting spring)	V_{el}	2	0.43	0.016
High-speed, with inertial pump (prototype VB-3T)	VI	2	0.41	0.004

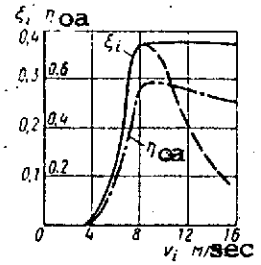

Fig. 27. Change in coefficient of efficiency and coefficient of energy utilization of a wind machine (solid line -- calculated values; broken line -- theoretical values.

$$A_{\Sigma o} = A_{o\ max} T_s \text{ kW·h} \qquad (92)$$

The output is calculated with various values for D_{wm}, v_{reg} and $v_{work\ max}$. The characteristics of the windmill and the load clearly define the relationship of v_0 with D_{wm}.

To transfer from the maximum possible output $A_{o\ max}$ according to wind conditions to the actual possible output A_{ao}, it is necessary to consider the coefficient of operation k_o which evaluates the simple machine according to technical and other reasons which are not connected with the lack of wind and the coefficient of coincidence k_c of the graphs of output and load (consumption), which is equal to 1 with a very high reservoir capacity and for wind-electric stations, working in parallel with a network. /54

TABLE 7. MAXIMUM VALUES OF ξ_i AND $\eta_{oa\ i}$ OF WATER LIFTING WIND MACHINES

Type of machine	H in m	Wind velocity in m/sec																	
		v_0		v_0+1		v_0+2		v_0+3		v_0+4		v_0+5		v_0+6		$v_{n.reg}+1$ h.reg & higher			
		ξ_i	η_{oa}	ξ_i	η_{oa}	ξ_i	η_{oa}	ξ_i	η_{oa}	ξ_i	η_{oa}	ξ_i	η_{oa}	ξ_i	η_{oa}	ξ_i	η_{oa}	ξ_i	η_{oa}
I	20	0,28	—	0,3	0,22	0,28	0,19	0,26	0,18	0,25	0,195	0,24	0,21	0,24	0,216	0,24	0,23	0,3	0,23
	30	0,28	—	0,3	0,18	0,29	0,17	0,28	0,2	0,27	0,24	0,26	0,265	0,25	0,28	0,25	0,29	0,3	0,29
II	20	0,3	—	0,33	0,488	0,29	0,51	0,27	0,538	0,24	0,58	0,22	0,605	0,21	0,675	0,2	0,675	0,33	0,675
	30	0,31	—	0,33	0,495	0,31	0,516	0,3	0,55	0,29	0,595	0,27	0,618	0,26	0,675	0,25	0,675	0,33	0,675
III	20	0,37	—	0,38	0,144	0,36	0,133	0,35	0,129	0,33	0,152	0,31	0,148	0,29	0,153	0,27	0,153	0,38	0,153
	30	0,37	—	0,38	0,131	0,37	0,165	0,36	0,191	0,34	0,178	0,32	0,183	0,3	0,188	0,29	0,188	0,38	0,188
IV	20	0,37	—	0,4	0,088	0,37	0,1	0,34	0,12	0,32	0,133	0,29	0,149	0,26	0,159	0,24	0,158	0,4	0,158
	30	0,38	—	0,4	0,069	0,38	0,074	0,36	0,088	0,35	0,1	0,34	0,116	0,32	0,14	0,3	0,14	0,4	0,14
V_{pn}	20	0,4	—	0,43	0,088	0,41	0,1	0,4	0,12	0,38	0,133	0,36	0,149	0,34	0,159	0,32	0,158	0,43	0,158
	30	0,39	—	0,42	0,069	0,4	0,074	0,37	0,088	0,35	0,1	0,34	0,116	0,33	0,14	0,33	0,14	0,43	0,14
V_{el}	20	0,4	—	0,43	0,059	0,41	0,077	0,4	0,087	0,38	0,096	0,36	0,1	0,34	0,117	0,32	0,14	0,43	0,14
	30	0,4	—	0,43	0,07	0,41	0,09	0,4	0,1	0,39	0,11	0,38	0,12	0,37	0,14	0,35	0,16	0,43	0,16
VI	20	0,4	—	0,41	0,6	0,4	0,8	0,39	0,6	0,38	0,42	0,37	0,33	0,39	0,2	0,35	0,2	0,41	0,2

$$A_{ao} = A_o \max T_s k_o k_c \ \text{kW·h} \tag{93}$$

The vertical profile of wind velocity in the surface stratum is determined according to formulas (6) and (7), and then repetition of velocities is recalculated for different heights (see Table 1).

The minimum working velocity of the wind v_0 depends on a number of values. If the given moment is M_{ax} and the aerodynamic layout of the windmill is selected, from equation

$$M_0 = \overline{M}_0 \pi \frac{D_{wm}^3 \rho v_0^2}{8} \tag{94}$$

we obtain

$$v_0 = \sqrt{\frac{K_0}{D_{wm}^3}} \ \text{m/sec} \tag{95}$$

where

$$K_0 = \frac{16 M_{ax}}{\overline{M}_0 \pi \rho},$$

$$M_0 = M_{ax}.$$

With D_{wm} = const, the value for $\bar{M}_0$ depends on the type and characteristics of the windmill. Velocity v_0 is computed for windmills of various speeds Z_n (the profile is selected, the number of blades i = var) and with various aerodynamic profiles (i = const). For windmills already completed, the dependency $\bar{M}_0 = f(i)$ is shown in Fig. 28.

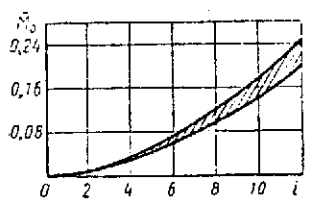

Fig. 28. Character of change relative to the initial moving moment of windmills with various numbers of blades (the zone of possible changes of moment $\bar{M}_0$ is streaked).

We present the weight of the machine as the sum of the weights of its components: of the windmill G_{wm}, the tower G_{to}, the head G_h, the orientation mechanism G_{or}, the water lift G_{wl}, and also of additional elements G_a:

$$G_{ma} = G_{wm} + G_{to} + G_h + G_{or} + G_{wl} + G_a \text{ kg}. \quad (96)$$

The basic weight is usually given by the first, second and third components and also the water lift; G_{to} and G_a practically do not depend on the base velocities of the wind and the height of the tower, and their effect on the value of G_{ma} with relatively small variations in the values of initial parameters can be ignored. If the range of changes in windmill diameters is significant, then as the result of an increase in the reach and area of the tail vane or the diameter of the wind rose, G_{or}, as statistical treatment of data showed, increases proportional to the relationship of the calculated and initial diameters in a degree of 0.33-0.5. The weight of the head does not depend on h_{to}, but depends on the diameter and type of the windmill. The weight of the pumping equipment depends on the type of the water lift and static pressure P_{st}. /55

The weight of the windmill is found according to the following formula or according to Fig. 29:

$$G_{wm} = G_{wm.in}\left(\frac{D_{wm}}{D_{wm.in}}\right)^{n_5} \quad (97)$$

or

$$wm = G_{wm.in}\left(\frac{v_{work\ max}}{v_{in.work\ max}}\right)^{n_6} \quad (98)$$

The values of exponents for low power machines are presented in Table 8.

The weight of the tower depends on its height, velocities of $v_{n.reg}$ and v_{max} and aerodynamic loads, determined by the diameter and type of the windmill. The effect of h_{to} on G_{to} is evident from Table 9 and Fig. 30.

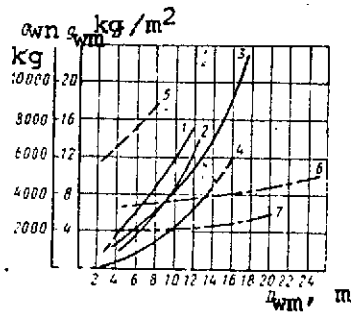

Fig. 29. Absolute and specific weight of windmills and wind engines of various types: 1. weight of low-speed engines with towers; 2. same, without tower; 3. weight of high-speed engines with towers; 4. same, without tower; 5, 6 and 7. specific weight of low-speed windmills and high-speed windmills types D-18 and 1D-18 respectively.

The change in weight of the support in the function $v_{work\ max}$ is determined according to the following formula

$$G_{to} = G_{to.in}\left(\frac{v_{work\ max}}{v_{in.work\ max}}\right)^{n_7} \qquad (99)$$

For intervals of height and diameters, the weight is determined with linear interpolation.

An increase in the vertical component of weight from the wheel and head increases G_{to} in the degree of approximately 0.5 from the relationship of the initial and sought weights. Weight G_h is affected little by initial velocities /56 of the wind; it depends on the construction of the head, and the weight and type of the windmill. For low-power engines, it can be considered that

$$G_h = G_{h.in}\left(\frac{D_{wm}}{D_{wm.in}}\right)^{n_8} \qquad (100)$$

and

$$G_{to} = G_{to.in}\left(\frac{D_{wm}}{D_{wm.in}}\right)^{n_9}. \qquad (101)$$

The cost of the machine depends on many factors: the scope, technology and organization of production, the weight and complexity of the components and the materials used. On the basis of statistical treatment of factual data on expenditures for materials and labor, plant cost and output price of the machine, the relationship between the weight of the component and its cost is established using the coefficient of cost p_{ci} and coefficient k_{ci} of adjustment to the cost of the base component (for single-type machines), on which the windmill is accepted, is determined. Since the construction and weight of the tower, orientation mechanism and auxiliary equipment are clearly not connected with the type of the machine, for these components $p_{ci} = 1$ is used. A machine of type I is used as the base. We note that the cost of manufacturing 1 kg of a component at various plants deviates from the average value within limits of /57 ±10%. Therefore,

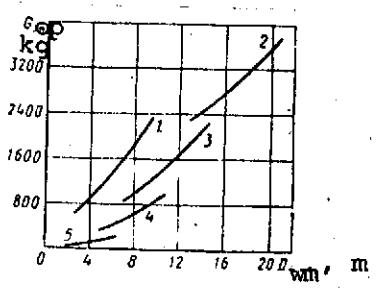

Fig. 30. Dependency of tower weight on its height: 1. low-speed engine, h_{to} = 15 m; 2. high-speed engine, h_{to} = = 20 m; 3. Same, h_{to} = 15 m; 4. Same, h_{to} = = 10 m; 5. Same, h_{to} = 5 m.

$$C_{ma} = k_{c_1} p_{c_1} G_{wm} + k_{c_2} G_{to} + k_{c_3} p_{c_3} G_a + k_{c_4} G_{or} + + k_{c_5} p_{c_5} G_{wm} + k_{c_6} G_a \text{ rub} \quad (102)$$

§ 4. Program for Solving Problems on Optimization

Let us suppose that it is necessary to find a machine for pumping water with the most economical parameters for set conditions. The pressure with which it will work and how much water is necessary to lift over the calculation period are known. It is not difficult to compute the quantity of energy which is required to perform this work, assuming at first that $\eta_{oa} = 1$.

$$E = \frac{Q_\Sigma H_\Sigma \gamma}{75 \cdot 1.36} \text{ kW·h} \quad (103)$$

For pasture watering points, these data are presented in Table 10.

TABLE 8. EXPONENTS FOR COMPUTING WEIGHT OF WIND MACHINE COMPONENTS (WITH $v_{n.reg}$ = 8 m/sec, $v_{work\ max}$ = 20 m/sec v_{max} = 30 m/sec)

Type of machine	Exponents				
	n_0	n_1	n_2	n_3	n_4
I	2,3	1.7	1,2	0.33	0,33
II	2,15	1,7	1.2	0.33	0.33
III	2,6	1,65	1.2	0,33	0.5
IV	2.7	1.6	1,25	0.25	0,5
V	2,7	1.6	1,35	0.25	0.25
VI	2,5	1,5	1,4	0.2	0.33

TABLE 9. WEIGHT INDICES OF WIND ENGINE TOWERS

Type of machine	Windmill diameter in m											
	2	3	4	5	3	5	7	4	6	8	5	8
	Weight of tower in kg with height in m											
	5				7			10			15	
I	60	90	110	200	175	300	500	360	570	900	1200	1900
II	80	112	135	190	185	330	530	360	570	900	900	1200
III—VI	50	70	100	160	140	250	410	270	430	660	700	1000

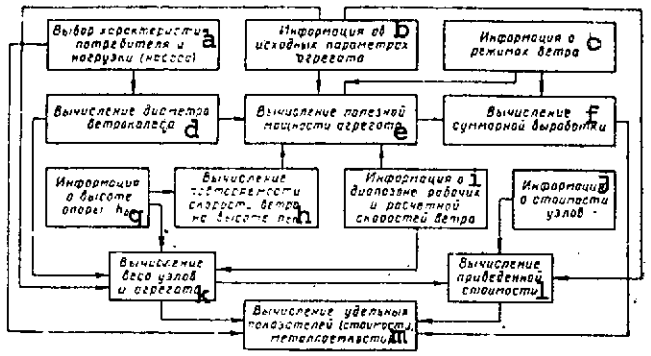

Fig. 31. Consolidated block diagram of solution of a problem in optimization of parameters of a wind machine.

Key: a. Select characteristics of demand and loading (of the pump); b. Information on initial parameters of machine; c. Information on wind rates; d. Compute diameter of windmill; e. Compute usable power of machine; f. Compute total output; g. Information on height of support h_{spt}; h. Compute repetition of wind velocity at h_{spt}; i. Information on spread of working and calculated wind velocities; j. Information on cost of components; k. Compute weight of components and machine; l. Compute adjusted costs; m. Compute specific indices (cost, metal consumption)

Therefore, we have the information on energy. For each working rate of the machine, the lifting of water goes on at different values for the coefficient of efficiency, which is taken into account by the proper expressions.

The principal sequence in solving problems on optimization is clear from the consolidated block diagram (Fig. 31) and from a variation of the machine program for the "Minsk-2" computer (Fig. 32). The program is relatively flexible and allows computation of many variations without regenerating initial data. It provides the /60 possibility for determining the parameters characterizing the machine and the conditions for its operation, to compute its total output, compare it with calculated values and reject variations which do not satisfy the requirements. With this, the quantity of calculated variations is fixed, and the optimal one is selected from them. As the parameters are selected, the weight of the components is also computed. Only those variations for which the values of output no lower than $0.9\ A_n$ are used in further computations; it is not limited according to the maximum $A_{w.w}$. An output which exceeds the required quantity of energy (Table 10) is replaced with the nominal value A_n. The limiting indices relating to nominal and possible outputs are in this case equal to each other.

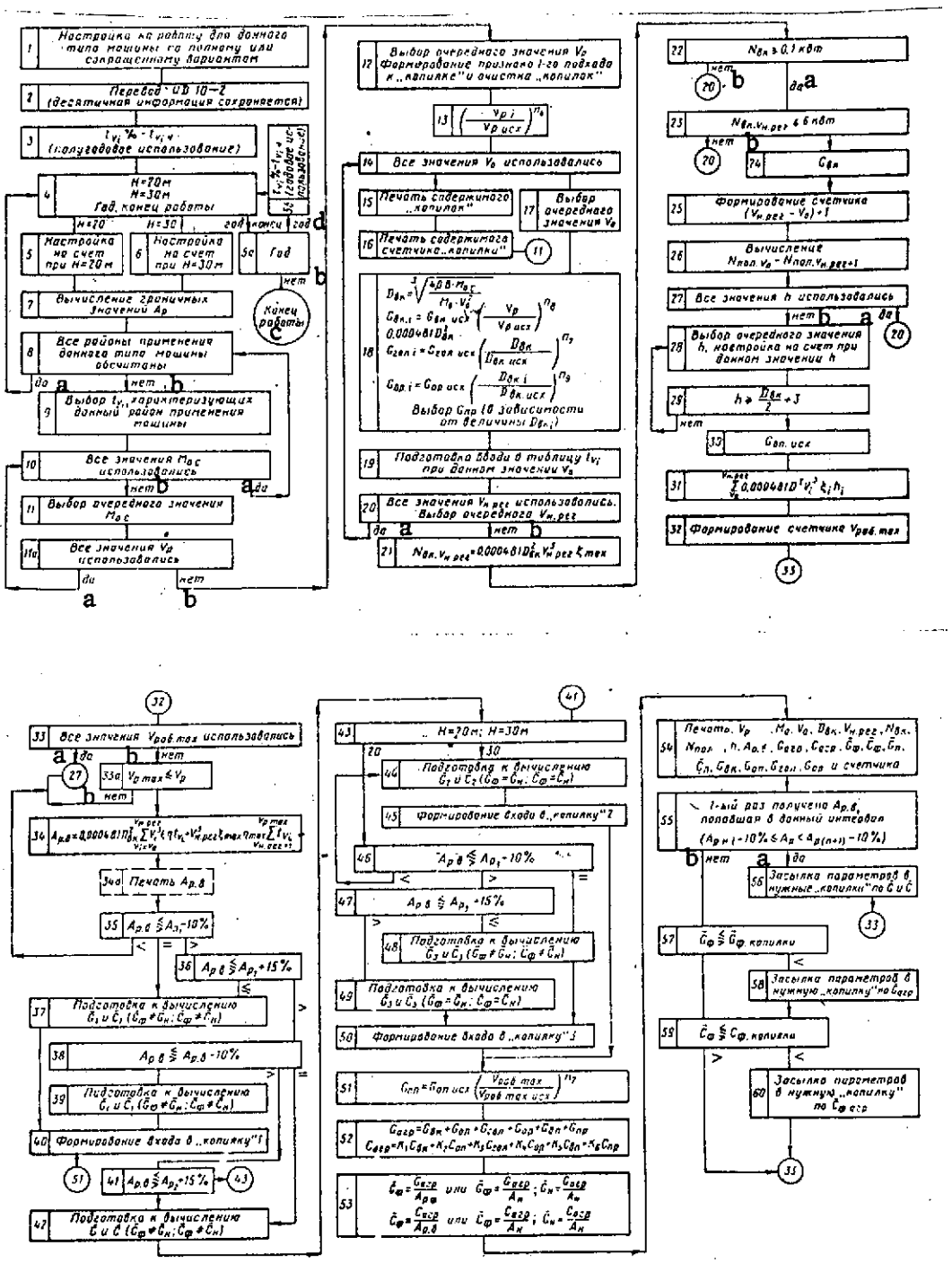

Fig. 32. Machine program for solving problems on optimization of parameters ("Minsk-2" computer).

Fig. 32: Key:

a. Yes
b. No
c. End of program
d. Year End Year

1. Set-up for operation for given type of machine
2. Convert ID 10 → 2 (retain decimal data)
3. t_{vi} % - t_{vi} h semiannual use
4. End of yearly operation
5. Initialize calculations for H = 20 M

5a. Year

5b. t_{vi} % - t_{vi} h annual use

6. Initialize calculations for H = 30 M
7. Calculate extremal values A_c
8. All regions of use for given type of machine have been calculated
9. Select t_{vi} characterizing given region of use of the machine
10. All values M_{or} were used
11. Select current value of M_{or}

11a. All values V_c were used

12. Select current value of V_c
    Set up criterion for first approach to register and clear registers
13. $(V_{ci}/V_{c\ init.})^{\pi}6$
14. All values V_o were used
15. Print contents of registers
16. Print contents of counter register
17. Select current value of V_o
18. $$D_{wm} = \sqrt[3]{\frac{4.08\ M_{or}}{M_o \cdot V_o^2}}$$
    $$G_{wm.i} = G_{wm\ init}(V_c/V_{c\ init})^{n8}$$
    $$0.000481\ D_{wm}^2$$
    $$G_{hd.i} = G_{hd.init}(D_{wm}/D_{wm\ init})^{n7}$$
    $$G_{or\ i} = G_{or\ init}(D_{wm\ i}/D_{wm\ init})^{n9}$$
    Select G_{dr} (depending on value of $D_{wm.i}$
19. Prepare input for t_{vi} table for given value of V_o
20. All values $V_{n.reg}$ were used. Select current $V_{n.reg}$
21. $N_{wm}V_{n.reg} = 0.000481\ D_{wm}^2 V_{n.reg}^2 \xi_{max}$
22. $N_{wm} \geq 0.1$ kW
23. $N_{wm} \leq 6$ kW
24. G_{wm}
25. Set up counter $(V_{n.reg} - V_o) + 1$
26. Calculate $N_{tot.V_o} - N_{tot.V_{n.reg+1}}$
27. All values of h were used

28. Select current value of h, initialize computations for given value of h
29. $h \geq (D_{wm}/2) + 3$
30. $G_{or.init}$
31. $V_{n.reg}$ $\sum_{V_o}$ $0.000481\ D^2 V_i^3 \xi_i h_i$
32. Set up counter $V_{work\ max}$
33. All values $V_{work\ max}$ were used
33a. $V_{c\ max} \leq V_c$
34. $A_{cw} = 0.000481\ D_{wm}^2 \sum_{V_i \geq V_o}^{V_{n.reg}} V_i^3 \xi_{nt_{v_i}} + V_{n.reg}^3 \xi_{max} n_{max} \sum_{V_{n.reg+1}}^{V_{c\ max}}$
35. $A_{cw} \lessgtr A_{v_i} - 10\%$
36. $A_{cw} \lessgtr A_{c_i} + 15\%$
37. Get ready to calculate G_1 and C_1 ($G_f \neq G_n; C_f \neq C_n$)
38. $A_{cw} \lessgtr A_{c,w} - 10\%$
39. Same as 37
40. Set up input for register 1
41. $A_{cw} \lessgtr A_{cw} + 15\%$
42. Get ready to calculate G and C ($G_f \neq G_n; C_f \neq C_n$)
43. H = 20 M; H = 30 M
44. Get ready to calculate G_2 and C_2 ($G_f = G_n; C_f = C_n$)
45. Set up input for register 2
46. $A_{cw} \lessgtr A_{c_1} - 10\%$
47. $A_{cw} \lessgtr A_{c_3} + 15\%$
48. Get ready to calculate G_3 and C_3 ($G_f = G_1; C_f = C_1$)
49. Get ready to calculate G_3 and C_3 ($G_f = G_1; C_f = C_1$)
50. Set up input for register 3
51. $G_{tr} = G_{tr\ init}(V_{work\ max}/V_{work\ max\ init})^{n_7}$
52. $G_{mach} = G_{wm} + G_{tr} + G_{hd} + G_{or} + G_{wl} + G_{dr}$
$C_{mach} = K_1 C_{wm} + K_2 C_{tr} + K_3 C_{hd} + K_4 C_{or} + K_5 C + K_6 C_{sp}$
53. $\bar{G}_a = (G_{mach}/A_{ca})$ or $\bar{G}_a = (G_{mach}/A_n)$; $G_n = G_{mach}/A_n$
54. Print V_c, M_o, D_{wm}, $V_{n.reg}$, N_{wm}
N_{tot}, h, A_{ca}, G_{mach}, C_{mach}, $\bar{G}_a$, $\bar{C}_a$, $\bar{G}_n$, C_n
G_{wm}, G_{tr}, G_{hd}, G_{or} and counter
55. First time the A_{cw} obtained lies in the given interval (A_{cni} $-10\% \leq A_{c(n+1)} - 10\%$)
56. Store parameters in appropriate registers according to G and C
57. $\bar{G}_a \lessgtr \bar{G}_a$ in register
58. Store parameters in appropriate register according to G_{mach}
59. $\bar{C}_a \lessgtr C_a$ in register
60. Store parameters in appropriate register according to $C_{a\ mach}$

TABLE 10. REQUIRED QUANTITY OF POWER IN kWh.

Period of Utilization of Machine	Number of Hours of Use	Consumer				
		One Flock of Sheep		Two Flocks of Sheep		Two Flocks of Sheep and Melon Field
		H in m				
		20	30	20	30	20
Season (6 months)	4380	80	120	160	250	500
Year	8760	160	240	320	500	1000

§ 5. Effect of Parameters on Effectiveness of Wind Machines

There are a number of dependencies relating to parameters of wind machines whose qualitative character can, to one degree or another, be presumed beforehand. However, quantitative interrelationships of the parameteters and the effect of specific features of the machine on its economy can be established only with calculations and with comparison of variations. We will look at this effect in an example of pumping machines, used in areas with average periodical wind velocities $\bar{v}_{per}$ = 3-7 m/sec, with the most characteristic type of velocity distribution:

In regions with $\bar{v}_{per}$ = 3-4 m/sec

$$t_{v_i} = 1258 \frac{\Delta v}{\bar{v}_{per}} \left(\frac{v_i}{\bar{v}_{per}}\right)^{0,45} e^{-0,867\left(\frac{v_i}{\bar{v}_{per}}\right)^{1,45}} \text{ h;}$$

In regions with $\bar{v}_{per}$ = 5-7 m/sec

$$t_{v_i} = 1571 \frac{\Delta v}{\bar{v}_{per}} \cdot \frac{v_i}{\bar{v}_{per}} e^{-0,785\left(\frac{v_i}{\bar{v}_{per}}\right)^2} \text{ h.}$$

The economy of the machine is increased in zones where the wind rate is more intensive. Thus, for instance with $\bar{v}_s$ = 5 m/sec, expenditures for lifting water with a type I machine are 1.5-2 times lower than in zones where $\bar{v}_s$ = 3 m/sec. For type III machines, with an increase in $\bar{v}_{per}$ by 1 m/sec (from 6 to 7 m/sec) expenses are decreased by 25-30%. The economy of type II machines depends less on velocity $\bar{v}_{per}$: the difference in the value of $\bar{E}$ for zones with $\bar{v}_{per}$ = 3 and 5 m/sec does not exceed 15-25%. As a rule, the higher the speed of the machine, the stronger the effect of the wind rate is felt on its economy. In this respect, there are sufficiently evident conformities which must be considered when founding recommendations on regional usage of machines. /61

The period of utilization of the installation and the quantity of energy required have the most essential influence on economic indices. In year-round pasture, the adjusted expenses, with otherwise identical conditions, are 1.8-2.3 times lower than in seasonal ones. At the same time, with an increase in water demand by double, expenditures $\bar{E}$ decrease by an average of 30-40%.

Moment M_{ax} should not in all cases be decreased, since, although starting characteristics are improved, the economy of the machine does not always increase. A clear relationship between M_{ax} and $\bar{C}_{ma}$ is absent and there is a zone of optimal values of M_{ax} in which the economy of the machine is the maximum. This is explained by the fact that for a given type of machine, the diameter of the windmill, in connection with the value of M_{ax}, determines the output and cost of the machine and therefore also determines expenses $\bar{E}$. Consequently, it is necessary to select the gear ratio of the drive such that the moment of resistance transferred to the main shaft has a set value.

We will examine some interdependencies of parameters which are characteristic for machines.

Machines of types I and III have the feature that a change in moment M_{ax} usually has little effect on their indices; only in a number of cases in a zone of high values of M_{ax} does the effectiveness of the machine become higher.

Optimum low-speed machines have D_{wm} = 3-5 m; h_{to} = 5-10 m, v_0 = 2-3 m/sec, and $v_{n.reg}$ = 7-9 m/sec. A change in v_{max} from 25 to 30 m/sec has practically no effect on their effectiveness.

The combination of the belt water lift with the medium-speed engine provides better results than combination with a low-speed engine, although it also leads to an increase in D_{wm} (expenses under comparable conditions are reduced by 30-70%). The economy of such machines is increased with an increase in their speed and velocity $v_{n.reg}$ up to 10 m/sec and a decrease in v_0 to 2 m/sec.

Usually the parameters of machines corresponding to the condition $\bar{E}$ = min and G_{ma} = min coincide and only in some variations are their divergences in the value for optimum tower height (Fig. 33).

The economy of low-speed machines with piston pumps depends little on the value of M_{ax}, and only in a zone of medium values where the moments (12-20 n·m) is a reduction in expenditures by 3-5% observed (Fig. 34). This is explained by two factors: the closeness of values $\bar{M}_0$ and $\bar{M}_{work}$ of a low-speed windmill and displacement of the v_0-$v_{n.reg}$ range toward higher values. For

higher values of M_{ax}, it is profitable to increase v_0, simultaneously narrowing the v_0-$v_{n,reg}$ range. By varying the gear ratio in the drive of a sucker-rod pump, it is possible to obtain an adjusted value for M_{ax} which corresponds to the minimum expenditures of the machine.

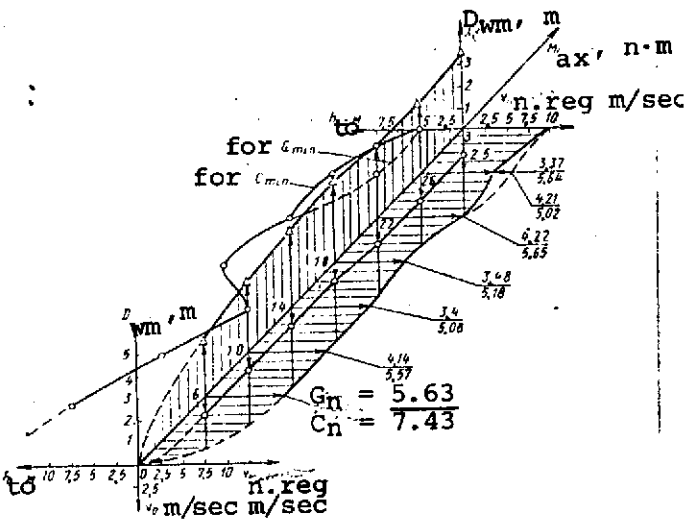

Fig. 33. The effect of base wind velocities on indices of low-speed machines with belt water lifts ($\bar{v}_s$ = 4 m/sec; H_Σ = 20 m; B_s = 6 m3).

The comparison of piston machines with low-speed belt ones shows the advantage of the former in zones with $\bar{v}_{per}$ = 3-4 m/sec, and with high demand for water also in zones with a more intensive rate. If conditions permit utilization of sucker-rod pumps, the use of low-speed machines should be recommended.

Pneumatic machines, made without feathering devices and mechanisms for resetting blades at angle $\phi = \phi_{opt}$, and equipped with piston compressors, have low effectiveness. They are sensitive to change in resistance in the system and therefore during the process of operation an increase in load decreases effectiveness of the wind engine. The machine also has a high metal consumption; in zones with $\bar{v}_{per}$= 5 m/sec and H = 20 m at a watering point with one flock of sheep it is necessary to install an engine with a large diameter windmill (D_{wm} = 9-11 m). Even the use of a membrane compressor with a lower value of M_{ax} than a piston one does not always essentially increase the economy of the machine. /63

Inclusion of a system for automatic relieving in the layout of a piston-free compressor essentially increases the economy of the machine. By comparison with machines of other types, this machine is profitable for use in zones where $\bar{v}_{per} \geq$ 4m/sec. The effect of even small changes in initial conditions and parameters, height of water feed, load and wind rate, should be included in peculiarities of this machine.

From a comparison of the specific indices of types II and IV machines (Table 11), it is evident that if the compressor is not unloaded during the starting period, the indices in a type IV machine will prove to be worse than in a type II machine.

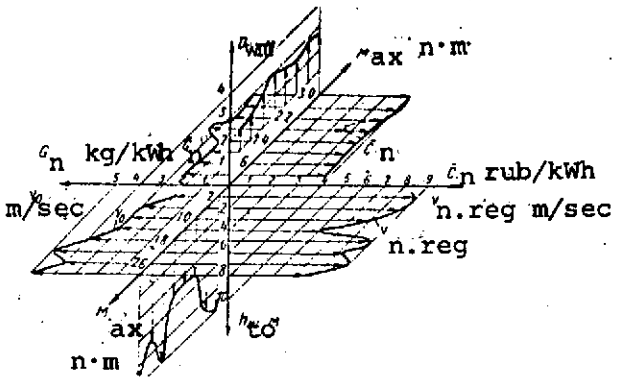

Fig. 34. Parameters of low-speed wind machines with piston pumps (v_s = 4 m/sec; H_Σ = 30 m; B_s = 6 m^3; solid lines -- v_{max} = 25 m/sec; broken lines -- v_{max} = 30 m/sec)

Relieving, leading to a decrease in the value of M_{ax} to 5 n·m, essentially improves the indices of the type IV machine, and for some conditions they prove to be higher than in type II machines. Since the specific metal consumption of a pneumatic machine is often lower, this should also be taken into consideration when developing type projects for utilization of wind installations. In particular, the initial introduction of pneumatic installations is expedient where there is a good wind rate, springs located in low places, and the water contains sand, which will lead to rapid wear in pumps of other types.

TABLE 11. COMPARATIVE DATA ON MACHINES OF TYPES II AND IV with $\bar{v}_c$ = 3 m/sec; v_{max} = 30 m/sec; H = 20 m; A_n = 80 kWh) /64

Type of machine	M_{ax} in n·m							
	6		10		25		30	
	$\bar{\sigma}_n$	$\bar{c}_n$	$\bar{\sigma}_n$	$\bar{c}_n$	$\bar{\sigma}_n$	$\bar{c}_n$	$\bar{\sigma}_n$	$\bar{c}_n$
II	4,12	5,70	3,76	5,7	4,36	5,91	4,38	5,93
IV	7,25	4,78	7,9	5,31	6,97	6,41	9,66	6,84

A type V pneumatic machine with a starting spring, providing its starting with $\bar{M}_0 = \bar{M}_{0\ max}$, and with a system for relieving, due to which $M_{ax} = M_{ax_{min}}$, is equipped with a membrane compressor, in which the moment M_{ax} is low, and the moment of loss due to friction comprises a small portion of the moment of total resistance. The economy of a type V machine, by comparison with a type IV machine, in zones with a moderate wind rate is greater by 15-20%, and with high water demand, in areas with $\bar{v}_{per}$ = 6-7 m/sec, it is higher by 25-40%.

The utilization of optimal starting rates increases economy of the type V machine by 25-30% (Table 12). This method of starting should be recommended for the majority of machines. The

introduction of relieving devices decreases expenses by 18-27% at the same time that the cost of the installation does not exceed 1.5% of the cost of the machine.

TABLE 12. INDICES OF TYPE V PNEUMATIC MACHINES (WITH ANNUAL WATER CONSUMPTION OF 2.2 THOUSAND m^3; H = 20 m; v_{max} = 30 m/sec; M_{ax} = 6 n·m)

(a) v_a в м/сек	(b) Диаметр ветроколеса в м		(c) Отношение показателей при обычном и оптимизированном пуске	
	(d) при обычном пуске	(e) при оптимизированном пуске	(f) удельный по весу	(g) удельной стоимости
3	6,71	5,54	1,32	1,33
4	5,14	3,94	1,36	1,33
5	5,14	3,94	1,78	1,45

Key: (a) v_a in m/sec
(b) Diameter of windmill in m
(c) Relationship of indices with normal and optimized starting
(d) With normal starting
(e) With optimized starting
(f) Specific weight
(g) Specific cost

An important aspect of pneumatic machines is the fact that their economic indices depend little on the value of v_{max} taken. This provides the possibility for increasing reliability of the engine without essentially increasing its cost. An increase in optimal values of calculated velocities, primarily $v_{n.reg}$ (usually 8-10 m/sec) is characteristic for them.

High-speed machines with electrical drive for centrifugal pumps usually have lower indices of effectiveness than machines of other types, even in cases where devices optimizing the rates of starting and acceleration are used. Only in zones with high values of v_{per} at objects with a daily water consumption over 15 m^3 do expenses on lifting water using such machines become equal to or even somewhat lower than expenses with utilization of machines having belt lifts. This is explained not only by the high initial cost of the machine with electrical drive, but also its low values of $\eta_{0\ all}$ due to the double conversion of energy (mechanical-electrical-mechanical) and a sharp change in the characteristics of the pump with $v_0 < v < v_{n.reg}$. /65

In solving problems connected with the use of electric pumping machines, a great influence is played by their operational advantages, including the possibility for assembling the engine at a significant distance from the water source, in a place which is elevated and favorable for free passage of the airflow. This often allows the total output of the machine to be increased by 15-20%, and thereby decreases specific expenses. An advantage of electrical pumping machines is also the fact that energy developed by the generator is easy to use for charging storage cells and for other purposes. This increases the value of k_e. Improvement of indices in machine with electrical drive is possible, by disconnecting the generator from the windmill during the entire period of starting and acceleration, adjusting the external characteristic of the pump (i.e. decreasing its transconductance in the zone of working velocities), and by using generators with a low initial moment of resistance. An important feature of machines with electrical drive consists of the fact that even small increases in v_{max} have an effect on their parameters and economy.

Machines of type VI with an inertia pump have a number of features with respect to their principle and construction. Thus, in a BV-3 machine, $v_{n.reg}$ = 6 m/sec, and not 7-8 m/sec as in machines of other types, while at the same time the maximum depth for lifting water is limited to 15 m. These machines are extremely effective. The range of use is v_0 = 5-7 m/sec, and $v_{n.reg}$ = = 6-9 m/sec; the difference between $v_{n.reg}$ and v_0 does not usually exceed 3 m/sec (for machines of other types, it reaches 5 m/sec) with the exclusion of zones with $\bar{v}_{per}$ = 3-4 m/sec, where this difference can reach 6 m/sec. The effect of loading on the diameter of the windmill is also increased, and the height of the tower (usually 5-7 m) and the effect of maximum speed on criterion of optimality are reduced.

A change in the cost of energy for optimal machines (Fig. 35) /66 shows that for providing water for two flocks with v_a = 5 m/sec, the most effective is a type VI vibration water lift (M_{ax} = 2-10 n·m), and for H > 20 m, type III machines. Cost indices of types I, II, III and VI machines depend little on initial characteristics of loading, which, in turn, affect the optimum diameter of the windmill. A transfer to the same conditions for seasonal utilization of machines increases expenses (for instance, for type III machines, by 21%). With this, the effect of M_{ax} is more sharply felt (Fig. 36).

When designing machines, it is important to pay attention to agreement of the aerodynamic characteristics of the engine with the initial moment of loading. Varying the transmission relationships (parameters) of the drive it is possible to attain minimal (for a given type of machine) values of $\bar{E}$.

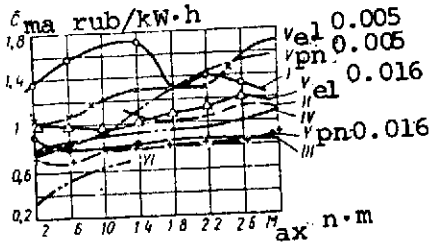

Fig. 35. Change in specific cost of energy of optimal wind machines used for providing water to two flocks of sheep over the course of a year ($\bar{v}_a$ = 5 m/sec; H_Σ = 20 m).

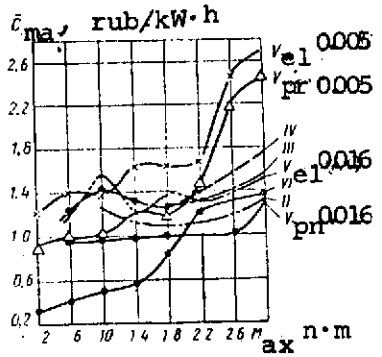

Fig. 36. Effect of loading on indices of wind machines for their season utilization ($\bar{v}_s$ = 5 m/sec; H_Σ = 20 m; $Q\Sigma_c = Q\Sigma_{max}$).

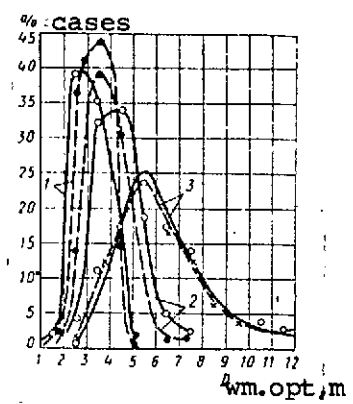

Fig. 37. Density of distribution of windmill diameters in water lifting machines: 1. low-speed, 2. medium-speed, 3. high-speed (solid lines -- annual utilization of machines; broken lines -- semiannual utilization).

As the results of optimization show, the most important factors determining effectiveness of an installation are the wind rate of the region and the periodicity of its utilization.

The values of $\bar{C}_{opt}$, each of which corresponds to optimal values of basic parameters, change within wide limits. For set conditions, the type of machine with $\bar{C}_{n\ min}$ is selected, and then values of the basic parameters are located. For instance, in regions with $\bar{v}_a$ = 5 m/sec, an inertia water lift, which is used over the course of a year for providing water to two flocks of sheep and irrigating a field of melons requires 3.5-5.3 times lower expenditures than machines of other types. With utilization of this water lift of seasonal pastures, expenses, by comparison with other machines, can be reduced by 2.8-4 times. /67

Many watering points function 5-6 months per year, which naturally brings about a number of additional requirements for means of mechanization; in particular, they must be more transportable, less metal consumptive and cheaper so that under these conditions they will be capable of being competitive by comparison with, for instance, heating installations. In conditions which are

characteristic for the central belt of the RSFSR ($\bar{v}_a$ = 4-4.5 m/sec, depth of water source is 20 m, head count at one point is 1500-2000 sheep), types VI, III and V_{pn} machines can be effectively used.

Distribution curves of windmill diameters for optimal machines change essentially depending on Z_n, displacing to the right and becoming more slanting as speed increases (Fig. 37). This indicates that the range of change in diameters with an increase in Z_n is somewhat broadened and a larger number of types of optimal high-speed machines might be demanded by comparison with low-speed machines.

Chapter 5. Wind Machine Layouts

The principal layout of a machine is dictated by its purpose, the type of windmill and characteristics of the machine's work. The rate and structure of the wind in the given region, economic and hydrogeological conditions, and peculiarities in the drive also have an effect on selection of the layout. Only by considering these factors is it possible to achieve the optimal characteristics of an energy-producing wind machine, it high reliability and its effectiveness. With this, the capability of the working machine and the device as a whole to work at a constantly changing velocity must be considered, as must the functioning stability of the drive, the effect of being not stationary and the dynamic processes occurring in the wind stream on the character of the technological process of accomplishing the machine and so forth.

Consequently, selection of the layout and its kinematic elements is one of the most important stages in creating the installation, whose optimal fulfillment increases effectiveness and reliability of the installations, and also eases its operation. In examining these processes, it is convenient to use the structural diagram shown in Fig. 38. From the diagram it is evident that between the working organs, i.e. the windmill, and the performing organ, i.e., the working machine (water lift, generator, etc.), there is only a transmission (drive), which can consist of mechanisms connected to each other by means of mechanical, electrical, pneumatic or hydraulic devices or their combinations (mixed drive). The remaining installations or elements of a wind machine as a rule fulfill an auxiliary role, during which many of them do not participate directly in transmitting the moment or power from the working organ to the performing one. The devices outlined by rectangles are mandatory for the machines, and those inscribed in circles are, as a rule, necessary. The remaining mechanisms are used depending on the actual layout and may not be present.

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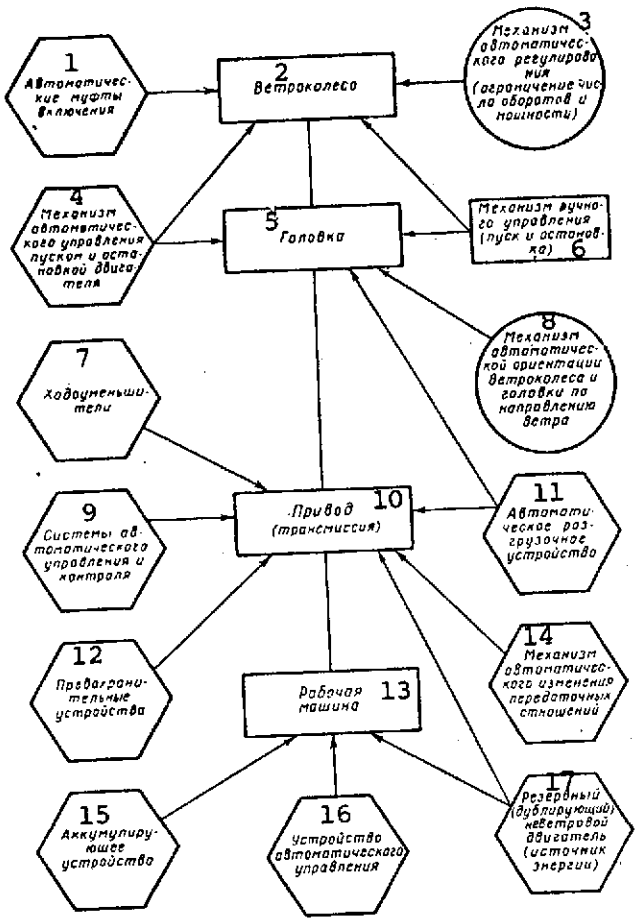

Fig. 38. Structural diagram of wind energy-producing installation.

Key: 1. Automatic clutches; 2. Windmill; 3. Automatic regulating mechanism (limiting the number of revolutions and power); 4. Mechanism for automatic control of starting and stopping the engine; 5. Head; 6. Manual control mechanism (starting and stopping); 7. Reduction gear; 8. Mechanism of automatic orientation of the windmill and head according to wind direction ;9. System of automatic control and monitoring; 10. Drive (transmission); 11. Automatic relieving device; 12. Protection devices; 13. Working machine; 14. Mechanism of automatic changing of drive ratio; 15. Storage cell device; 16. Automatic control device; 17. Reserving (duplicating) non-wind engine (energy source).

We will examine the most important types of machine layouts, classifying them according to the following groups: water-lifting, universal power, and wind-electrical.

§ 1. Layouts of Water-Lifting Machines

The majority of water-lifting machines, especially the earlier models, work with piston pumps or belt water lifts, whose characteristics necessitated the use of engines and corresponding types of drives. More often these were low-speed machines with mechanical drive, which make installation of the engine at a distance of more than 2 m from the water source difficult and increase the metal consumption of high- and medium-power machines. The following machines have the simplest layouts: those consisting of a low-speed windmill, connected through an upper crankshaft-connecting rod mechanisms with a sucker-rod pump; a high-speed windmill connected through an eccentric-connecting rod mechanism with a vibration (inertia) pump; a high-speed windmill with a pistonless compressor and pneumatic pump; a medium-speed pump connected through a mechanical

transmission to a belt water lift or to a screw-type pump, and a high-speed windmill with a generator which powers the magnetic device (vibrator) of a vibration pump. These machines, as optimization computations showed, are also the most effective.

Concerning the less economic wind-electrical machines with centrifugal pumps, when using them (primarily at large objects with high water consumption) it is necessary to take into account the fact that for great depths (H > 60 m) sufficiently effective water lifts of other types have not yet been created.

When using pumps of the rod type (piston, plunger), which accomplish a back-and-forth motion, the following machine layouts are most often used.

1. Pump drive is usually accomplished from a low-speed windmill through the crankshaft-connecting rod mechanism of the head. The rotational motion of the windmill is transformed into the motion of the rod. This layout (Fig. 39, a) is used in types TV-3, TV-5, and DDK-4 domestic pumping machines and in many foreign engines).

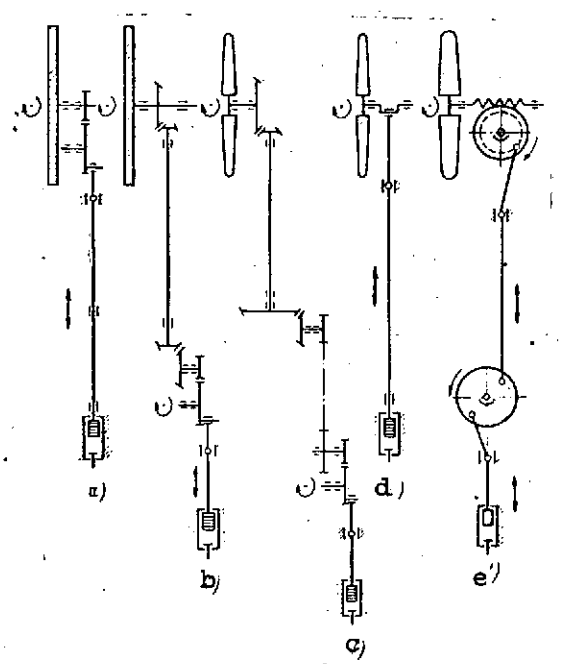

Fig. 39. Kinematic diagrams of machines with rod pumps.

2. The pump works from a universal windlass, driven into motion by the windmill through upper and lower reduction gears, which are connected with a vertical shaft (Fig. 39, b, c). The upper reduction gear of the head is conical or conic-cylindrical. The TV-8 (Fig. 39, b), UVD-8 and D-12 (Fig. 39, c) machines have these layouts. /70

3. Pump drive is accomplished immediately from a crank or crankshaft, onto which is fastened the windmill, usually a high-speed one (Fig. 39, d). This layout is used in the D-3 and VD-5.5 machines; foreign wind technicians use it rarely for reasons which are spelled out later. Evaluation of the principal layouts of machines is given in Table 13.

Regardless of the tendencies mentioned toward combination of wind engines with other types of water lifts, layouts of

TABLE 13. EVALUATION OF SOME PRINCIPAL LAYOUTS OF MACHINES WITH PISTON PUMPS

Diagram in Fig. 39	Advantages	Disadvantages
a	Simplicity of design execution; high value for overall coefficient of efficiency; stability of operation at low wind velocities	Necessity to install engine near the water source or above it
b,c	Universality and, in connection with this, the possibility for complex utilization; low values of minimum working wind velocities	Relatively high metal consumption; difficulty in assembly away from the water source (according to diagram b)
d	Low metal consumption; simplicity in design execution of the machine; high value for overall coefficient of efficiency	Low reliability of drive, especially at high wind velocities; stable operation begins at relatively high wind velocities.

machines with rod pumps, using higher speed windmills, introducing systems of relieving during the starting period and using more /71
modern transmission, double action pumps and plungers have been improved. In the PD-3M machine, the number of blades has been reduced to 12 (the TV-5 and TV-8 machines have 24 and 18 blades respectively); and an eccentric mechanism with a connecting rod and rocker arm are used in the head.

The PD-5M machine has a two-bladed windmill. For reducing the initial working velocities of the wind v_0, an automatic clutch is used in the transmission of the machine, and therefore the engine begins to work at v_0 = 3.5 m/sec and at $v_0 = v_{work\ min}$ = = 3.7 m/sec, the load is engaged. The use of this sytem allows the weight of the machine to be essentially decreased, but its construction is made more complex due to the presence of an additional lower worm reduction gear (windlass) and an automatic friction clutch. The latter, due to constant changes in wind velocity and, consequently, also frequency of rotation of the shaft, works unreliably, and upon engagement of the load, especially at time when the pump piston is located at bottom dead

center (BDC) or is moving upwards (pumping water), significant dynamic overloads arise in the transmission. Experience shows that for low-power specialized machines with piston pumps, the use of similar layouts, more complex construction and reduced coefficient of efficiency of the machine are most frequently expedient.

Besides using an automatic clutch, it is also possible to point to other means for improving the layouts and the characteristics of high-speed machines with rod pumps. The following should be included in these: /72

Increasing the initial aerodynamic moment of the wheel $\bar{M}_0$ by mean of widening blades and, consequently, decreasing nominal speed Z_n somewhat;

Automatic setting of blades at optimum angles during the starting and acceleration period of the machine, which increases $\bar{M}_0$ by 3-4 times;

Replacing the piston pump with a cup-free plunger one, having a lower initial moment of resistance;

Using a pump with a relieving valve or hole in the lower part of the main pressure line, through which water drains into the well upon stoppage of the engine, reducing part of the load on the piston, and the danger of water freezing is decreased during winter.

The first means is used in a machine proposed by S. L. Rozentul which has a cup-free pump (Fig. 39, e). Three wide blades allow attainment of $\bar{M}_0 = 0.5$, with which the value $\xi(Z)$ in the range of $Z = 2\text{-}5.5$ changes little (Fig. 40). The combination of a windmill with a high moment $\bar{M}_0$ and a pump with a relatively low initial moment of resistance M_{0r} reduces the value of minimum working velocities of the wind to 3 m/sec.

The utilization of a layout for direct pump drive from a high-speed windmill crankshaft does not provide positive results due to a decline in starting characteristics of the machine and high dynamic loads. The starting characteristics can naturally be improved if a windmill which is large in diameter by comparison with calculated diameter is used, but this increases metal consumption and decreases the effectiveness of the engine.

A layout with direct pump drive from the main crankshaft can be successfully used in cases where the water lift (pump) allows a large number of strokes per minute, and its initial moment of resistance is relatively low. This drive is used in the VB-3T machine with an inertia water lift, whose action is based on the

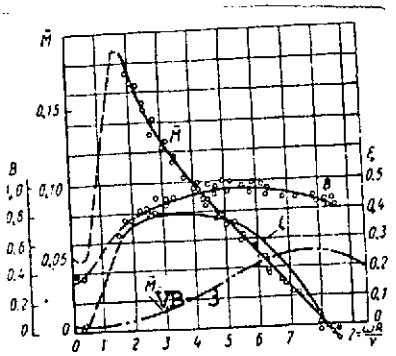

Fig. 40. Characteristics of a windmill with wide blades.

fact that with fluctuating motions of the water lift pipe occurring with low amplitudes (40-50 mm), but with relatively high frequency (8-10 Hz), the water is given the energy necessary for lifting it (see Fig. 52). /73

When using belt water lifts, layouts of two different types are primarily used:

1. High-speed or low-speed windmill connected with an upper conic or conic-cylindrical reduction gear, from which the belt is driven into motion through a vertical shaft and lower reduction gear with the water lift drive pulley. The layout is used in the VP-3M, VVU-3, VBL-3 and VTL-3 water pumping wind machines. The layout is relatively simple, although it has a deficiency in that the wind engine must be installed immediately at the head of the well.

2. A high-speed windmill drives into rotation the power pulley of the water pump, installed on the head of the well, through two identical, interchangeable conic reduction gears and a vertical shaft, using the Vee-belt drive.

This layout has a number of advantages. It allows utilization of a series-produced water pump. The engine is installed at a distance of 2-3 m from the water source, depending on actual conditions (depth and yield of the well, wind conditions and other); and the transmission ratio to the water pump is changed using removable pulleys. The layout is used on the "Buran" machine.

Water-lifting machines with rotary motion pumps: centrifugal impeller, screw-type and multicylinder plunger pumps with a slanted plate are more preferable. With a mechanical drive, the layouts of these machines have much in common.

The high-speed windmill is fastened onto the shaft of the reduction gear, from which the moment is transmitted to the vertical shaft. The drive rod of the pump is fastened to it. In some cases, a manual drive mechanism is mounted between the shaft and the rod, as is done in, for instance, the "Veterok" machine with a single-screw pump (see Fig. 67). During operation of the manual drive, the vertical shaft is disengaged using a dog coupling, and therefore it is not necessary to waste effort rotating the windmill.

The use of pneumatic drives and pumps provides the possibility for increasing the speed of the wheel, reducing the metal consumption of the machine, providing independence of the relative location of the engine and well, simplifying construction of the installation and increasing reliability of the pump.

The first pneumatic machines, created in the 1950's, were executed according to the diagram of engineer V. V. Savotin [54]. The VKU-3.5 installation had a double-bladed windmill (D_{wm} = = 3.5 m), a piston air compressor and a single- or double-chambered displacement pump. Compressed air was fed into the double-chambered pump through the slide valve type air distributor which was controlled by a float or pneumatic expander; a locking tube was used in the single-chambered pump. To ease starting conditions of the machine, the windmill hub has an automatic friction sleeve of the centrifugal type, which at $\omega_{wm} \geq \omega_{wm\ min}$ rigidly connects the hub with the compressor shaft. The deficiencies of the system are its complexity, and also the reduction in reliability and economic indices of the machine due to the use of a piston compressor and sleeve. /74

Further improvement of the layout led to creation of the VPV-5 machine, whose two-bladed windmill drives the shaft of the piston compressor into rotation through a chain drive and automatic shoe sleeve.

Analysis of the layouts and construction of pneumatic machines shows that their effectiveness and reliability can be essentially increased by not using centrifugal clutches, replacing the piston compressor with a pistonless one, and fitting the pump with an automatic device to provide the required air feed rates. Systems for relieving the compressor, controlling the centrifugal-aerodynamic regulator of the windmill, electromagnetic valve, wind regulator and others are provided to ease conditions of starting in the machine.

Of the pistonless compressors, it is advisable to use in wind machines diaphragm compressors, which have small moment M_{0r} and M_x and do not require lubrication of the working surfaces, therefore preventing lubricating oil from getting into the water.

An automatic device provides the possibility for adjusting the pump for operation at optimal rates under all pressures from minimum to calculated, provides an increased coefficient of efficiency and allows a reduction in the dimensions of the pump.

This feature is considered in the "Lastochka" and "Vikhr'" (Fig. 41) pneumatic machines.

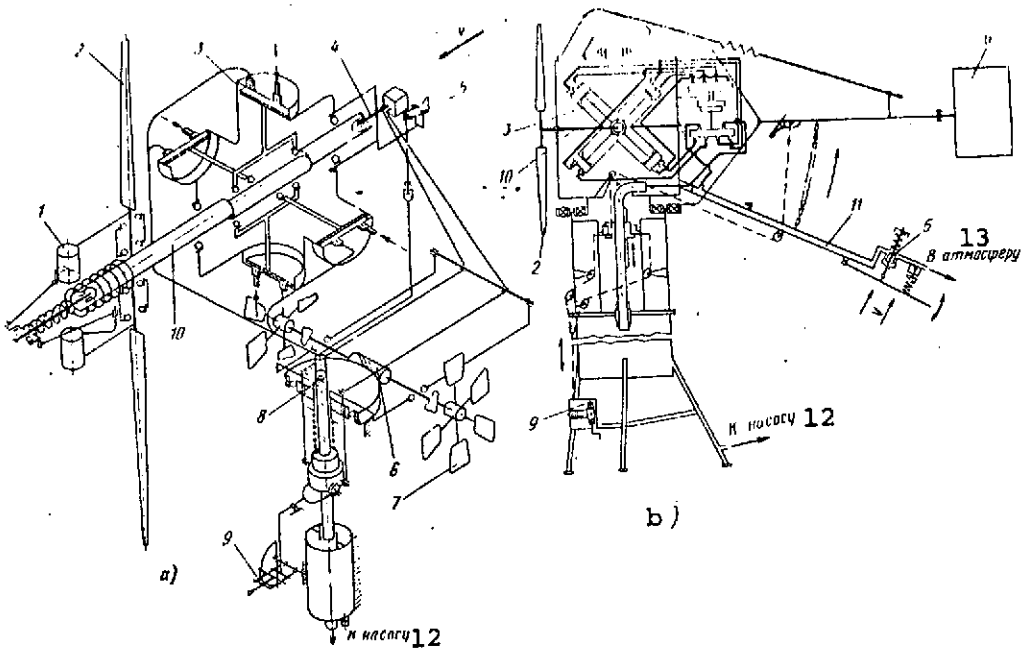

Fig. 41. Diagrams of high-speed pneumatic pumps of the machines: a. "Vikhr';" b. "Lastochka;" 1. Regulator; 2. Windmill; 3. Compressor; 4. Regulator rod; 5. Relieving device; 6. Orientation mechanism; 7. Windrows; 8. Check valve; 9. Starting and stopping windlass; 10. Main shaft; 11. Side plane; 12. To pump; 13. To atmosphere.

In development of wind-electric pumping installations, principal layouts of basically two types are used:

1) Combination of a high-speed windmill through an increased-speed transmission with a high-revolution generator and centrifugal pump;

2) Direct drive of a special low-speed generator from a high-speed windmill to supply energy to a standard pump motor.

The first layout possesses the advantage that it allows standard generators and pumps to be used. Using generators and motors of high frequency, by comparison with manufactured ones, increases the dimensions and weight of the equipment, and also increases stability of operation of the electrical drive. This layout is used in the "Berkut" machine. The second layout is simpler and provides the possibility for creating a machine which is more reliable in operation. It is used in the VES-1.5-6 machine, in which a generator (n_n = 230 rpm) with constant

magnets and a windmill (D_{wm} = 6 m) with a centrifugal regulator are used [46].

Comparing the two principal directions in creating electrical pumping machines and evaluating their positive and negative sides, /76
it is necessary to consider the operational and engineering-economic indices. With this, preference should evidently be given to machines created according to this first layout, whose economy is higher than those without reduction gears, and manufacture is simpler due to the utilization of a standard generator. However, the layouts without reduction gears, primarily in machines for pit wells, are interesting.

The combination of the motor support with the water pressure tower in a pumping wind installation is expedient. This problem is most simply solved in utilization of electrical or pneumatic drive, but can also be solved with a mechanical drive. In the latter case, the tower must be installed immediately over the water source.

§ 2. Layouts of Wind-Powered Machine

The layout for a wind-powered machine shown in Fig. 39, b and c is the most widely used. It is used in the TV-8 and D-12 motors, which over the course of many years have been produced serially and also in a number of installations produced by foreign firms. Depending on the system of regulating power and windmill rotation frequency used, the upper reduction gear is made to be single stage with a pair of conic gears or two-stage with one cylindrical pair and one conic pair. The drive of the working machines is accomplished from the pulleys of the lower reduction gear, usually single stage and, less frequently, two stage.

The deficiency of the layout is the presence of a long vertical shaft with a large quantity of intermediate supports which increases loss due to friction and metal consumption of the motor. The presence of a reaction moment deteriorates conditions of operation of the regulation system, which is operated by output of the windmill from the wind, and under changing loads causes additional oscillation of the head relative to the vertical axis, which increases gyroscopic loads. Therefore, during recent years, the wind-power machines with mechanical drive have not found widespread usage.

Electrical drive is beginning to be used in a complex of machines in the work of wind motors having a power of more than 4 kW. However, this causes a reduction in the overall coefficient of efficiency, but makes the machine more convenient in operation and less metal consumptive.

§ 3. Layouts of Electrical Wind Machines

The layouts of electrical charging wind machines (station) having a low power are similar to the layouts of pumping electrical machines. With a windmill diameter up to 3 m, preference is given to the layout without a reduction gear drive. When using high-speed generators, the use of two wind machines (propellers) rotating in different directions, one of which is connected with /77 the rotor and the other with the stator of the generator, also fastened on bearings, gives good results.

Wind-electrical machines and stations of medium and high power are usually designed according to a typical diagram: windmill-reduction gear-generator, with which all the basic elements (reduction gear, generator and others) are located on top and form the head. The "Sokol" machine with a moment-centrifugal system of regulation (Fig. 42) is, in particular, made according to this layout. The bodies of the reduction gear and generator in /78 this machine are combined in construction; this makes its head more compact.

In the 1D-18 TsAGI domestic wind-electric stations and also in the first lot of D-18 stations, the generator is located on the bottom and therefore two reduction gears and a vertical shaft are used. The advantage of this layout is simplified servicing and the possibility for the mechanical drive of machines from the lower reduction gear pulley. In the 1D-18 wind electric station, an inertia storage cell is used, and its location on the rotating head is made more difficult.

Other layouts, providing utilization of planetary reduction gears, super-slow generators or aerodynamic drives have not found widespread utilization and are used only in experimental models.

Chapter 6. Design of Wind Machines

During recent years, a number of new high-economy wind energy-producing machines of original design have been developed. At the same time, some machines created in the 1950's have not lost their value and continue to be effectively used in the national economy. Machines created in the years 1965-1970, having passed state testing and being produced or recommended for production are examined in this chapter. The layout of wind installations and wind electric stations developed earlier can be found in the literature [4, 14, 35, 46].

During creation of wind energy-producing machines, the designers strived first of all toward increasing the speed of the

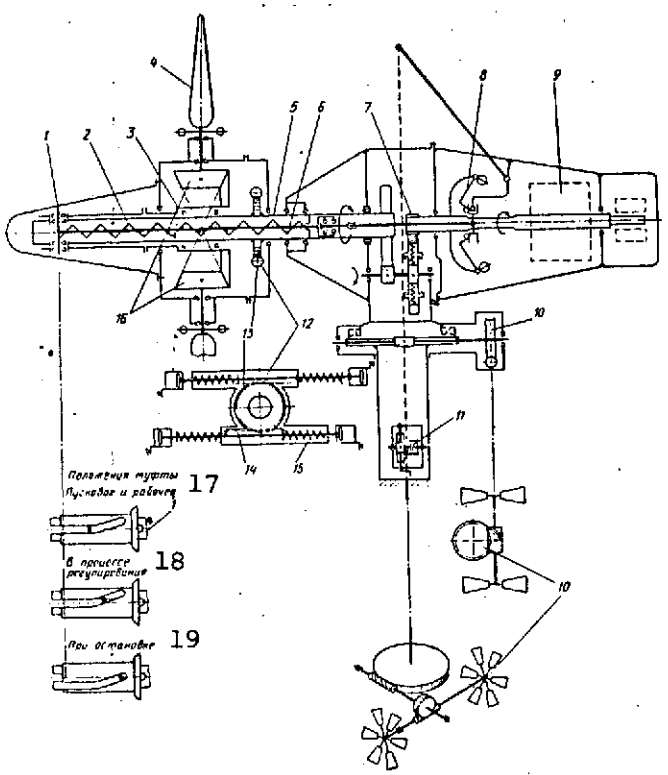

Fig. . 42. Diagram of the "Sokol" electrical wind machine: 1. Sleeve with slot; 2. Regulator spring; 3. Kinematic connection gear; 4. Windmill; 5. Main shaft; 6. Regulator rod; 7. Reduction gear; 8. Centrifugal regulator; 9. Generator; 10. Orientation mechanism; 11. Starting-stopping windlass; 12. Moment regulator; 13. Toothed wheel; 14. Toothed racks; 5. Moment regulator spring; 16. Stroke spring; 17. Starting and working positions of sleeve; 18. In the process of regulation; 19. When stopping.

windmills and unifying the basic components and mechanisms, primarily through the use of automatic systems of regulating the rotation frequency and limiting power by turning the blades. In the future, a gradual movement away from systems of regulation and the use of windmills with rigidly fastened blades possessing a property for aerodynamic self-adjustment are being proposed.

Widespread utilization of an automatic wind rose system of orienting the head with location of the wind rose in front of the windmill, introduction into the design of devices and mechanisms optimizing starting and acceleration rates, widespread utilization of aluminum alloys, low-alloy steels and nonmetallic materials, primarily fiberglass for the manufacture of blades, graphite and fluoroplast bushings for sliding bearings and prestressed reinforced concrete for towers are characteristic for modern machines. The method of grouping and assembly of machines out of consolidated components (blocks) assembled and adjusted under factory conditions, are ever more widely used, with which welding operations are excluded, since butt-joined elements are accomplished using bolts or other unified assembly connections. /80

The weight of each block (component) in a low-power machine has been reduced to 70 kg, in order to provide the possibility for assembly by two workers without using hoisting devices. The

reliable operation of the machine over the course of, as a minimum, a season is provided with only periodic servicing, without assembly, repair and replacement of parts, which is achieved by using enclosed tooth drives, roller bearings, one-time lubrication and others.

For pastures and wind-electric charging machines, a foundationless installation of the wind motor tower, having articulated devices easing its raising are most often used; a free-standing open tower is used primarily instead of a support on braces (guy wires) with which the engine is installed, as a rule, on the side of the water source.

Finally, for pumping and charging installations, a move away from using a duplicating nonwind engine for windless periods to provide a guaranteed water or electric supply is characteristic, due only to reserving of capacity or storage batteries, which, naturally, leads to increased requirements for strength of components and mechanisms, and for reliability of the entire machine. This position is to a greater or lesser degree reflected in designs of water pumping machines for pastures, low power charging wind-electrical machines and larger ones intended for work with electric motors of pumps and supplying electrical power for other consumers (Table 14).

§ 1. Water Lifting and Pumping Wind Machines

The VBL-3 high-speed machine with a belt water lift is calculated for lifting up to 3 m^3/h of water from a depth of 25-30 m and is designed for use in zones where $\bar{v}_a > 4$ m/sec. At an average wind velocity of 4-4.2 m/sec, the average daily feed of water is 11-12 m^3.

The machine (Fig. 43) consists of the windmill 1, head with regulator 2, tail 3, tower 4, lower reduction gear 5, band water lift 6 with guard 7, and starting and stopping mechanism 8.

The windmill (Fig. 44) has three blades one of high-strength, thin-sheet steel, which are set in all bearings 2 of hub 3 and can in the process of adjustment rotate relative to the longitudinal axis under the effect of the centrifugal forces of weights 4, the mass of the blade and working spring 5 of the regulator.

The position of the blade in the flow is determined by the frequency of rotation of wind wheel and by the wind velocity. When starting, due to spring 6, the angle of inclination comprises 30°, which gives a high starting moment and provides the beginning of operation of the wind with wind velocities of 3.5-4 m/sec. In /82 the working position, the angle of blade setting is equal to +7°, and upon stopping the blade is turned to a negative angle of -2°.

TABLE 14. TECHNICAL CHARACTERISTICS OF WIND ENERGY-PRODUCING MACHINES

	Indices	Machine												
		VDL-3	VTL-3	VB-3T	"Chayka"	"Bruan"	"Vikhr"	"Veterok"	"Berkut"	BE-2M	D-4	UVEU-(1)-4-6	VEV-1-6	"Sokol"
(a)	Диаметр ветроколеса в *м*	3	3	2,8	3	4	4	4	4	2	4	6	6	12
(b)	Число лопастей	3	18	2	12	3	2	3	2	2	2	2	2	3
(c)	Частота вращения ветроколеса на расчетном режиме в *об/мин*	270	90	420	61	108	320	270	300	600	280	186	250	88
(d)	Расчетная (номинальная) мощность ветроколеса в *квт*	1,0	0,8	0,7	0,75	1,6	1,6	1,6	1,6	0,15	1,6	3,4	3,2	15,2
(e)	Диапазон рабочих скоростей ветра в *м/сек*	4—35	3,3—18	3—40	3,5—17	4—17	4—40	3,5—40	4,5—40	3—25	4—40	4—40	4—30	4,5—40
(f)	Максимальная расчетная скорость ветра в *м/сек*	35	30	40	30	35	40	40	40	30	40	40	30	40
(g)	Общий к. п. д. агрегата при расчетном режиме	0,17	0,15	0,4	0,6	0,2	0,16	0,55	0,16	0,65	0,5	0,15	0 4	0,7
(h)	Расстояние оси ветроколеса от поверхности земли в *м*	5,75	5,8	6	5,1	5,1	5,1	5,1	5,1	7—12	5—7	9	6	10
(i)	Вес комплектного агрегата в *кг*	430	445	205	490	500	420	470	450	51	240	1300	1000	1900

Key: (a) Diameter of windmill in m
(b) Number of blades
(c) Frequency of windmill rotation at the calculated rate in rpm
(d) Calculated (nominal windmill power in kW)
(e) Range of working wind velocities in m/sec
(f) Maximum calculated wind velocity in m/sec
(g) Overall coefficient of efficiency of the machine at the calculated rate
(h) Distance of windmill shaft from the surface of the earth in m
(i) Weight of the assembled machine in kg

During the process of regulating the frequency of rotation and power, the angle of blade setting is changed within the range indicated (+7 to -2°).

The head (Fig. 45) consists of windmill rotation frequency regulator 1, single-stage conic reduction gear 2, support device 4, and starting and stopping mechanism parts, which include lever 3, clutch 5, enclosed by housing 6, cables and others. The

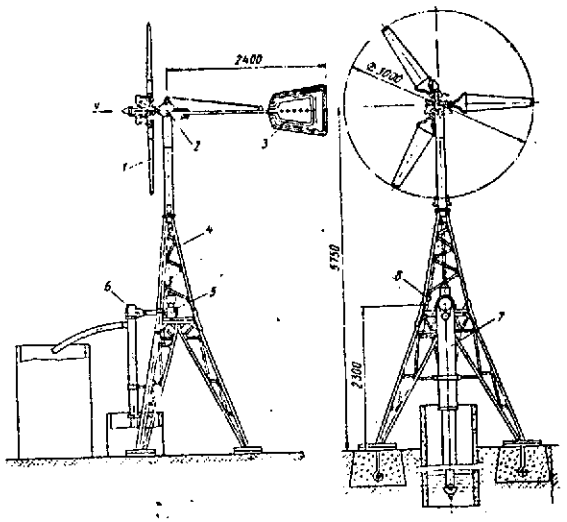

Fig. 43. Overall view of VBL-3 machine.

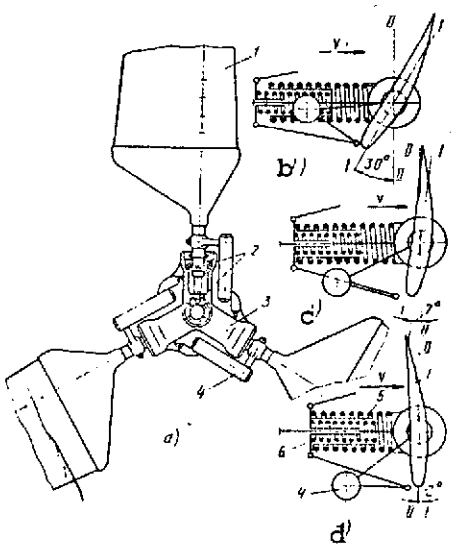

Fig. 44. Windmill and regulator diagram of the VBL-3 machine: a. Windmill; b, c, d. Starting, working and disengaging position of blade; I-I. Blade section chord; II-II. Plane of rotation of windmill.

transmission ratio of the reduction gear is 1 : 1.38. Self-compressing seals are used for sealing the space of the reduction gear and the support cup. A tail, whose empennage is deflected by 40° to the left, looking from the side of the windmill, is fastened to the head body.

The tower is a three-edged welded girder, assembled of upper and middle sections and three base supports, two of which have articulate fastening to the foundation plates, and the third, a device for installing a lifting lever. /83

The lower reduction gear (Fig. 46) has a conic toothed pair (i = 1.38) and cuff 1, which is terminated with drive pulley 2 of the belt water lift. To improve the operating conditions of shaft 4, they are connected to each other with elastic sleeves 3.

The water lift is an endless perforated rubberized belt on which the tension roller located inside a cast iron body hangs. The water is raised due to the forces of adhesion of particles to the material of the belt and is thrown into the housing in the upper part of the drive pulley under the action of centrifugal forces. From the housing the water flows into a tank along an outlet pipe.

Forced starting and stopping of the machine is accomplished with a lever, which is fastened on the lower part of the tower.

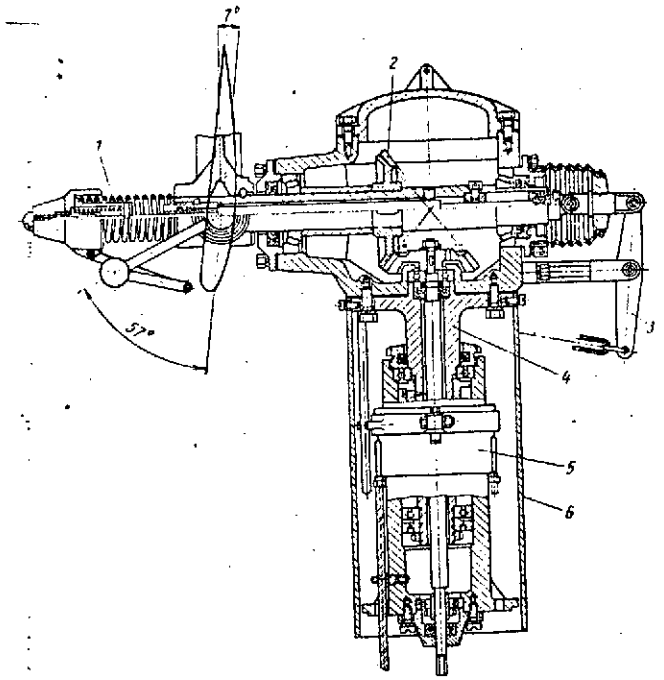

Fig. 45. Head of the VBL-3 machine.

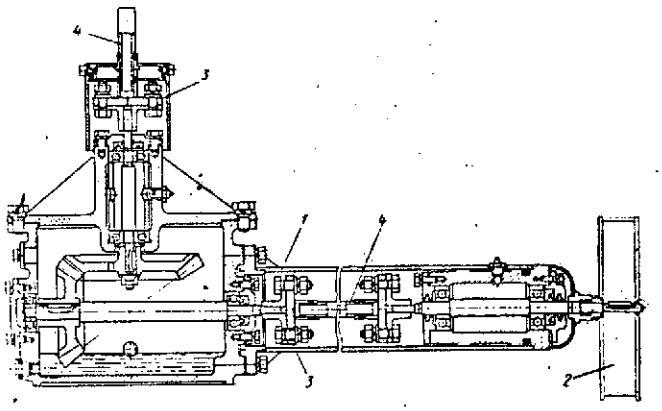

Fig. 46. Lower reduction gear of VBL-3 and VTL-3 machines.

The VTL-3 low-speed (Fig. 47) has a number of parts (tower, lower reduction gear, water lift), which are unified with similar components of the VBL-3 machine, but is equipped with a multi-bladed windmill. Therefore, it can also be effectively used in zones with a less intensive wind rate. With an average periodical wind velocity of 4-4.2 m /sec, the daily feed of water is 13-14 m^3.

The machine consists of seven basic components which are connected using bolts. The distinguishing feature of the windmill (Fig. 48), which has 18 blades, is the fact that the blades are fastened behind a load-bearing frame. This improves starting characteristics of the machine and reduces aerodynamic losses.

The head (Fig. 49) is a two-stage conic-cylindrical reduction gear (i = 3.78). Due to displacement of the windmill axis of rotation relative to the axis of rotation of the head, a limitation of the power of the windmill output due to the wind is accomplished. The machine is disengaged from operation at v_{max} = 18 m/sec. /85

The VB-3T high-speed machine with inertia water lift is designed for lifting water from pit andbored wells with a diameter from 100 mm with a depth of water location up to 15 m. It is used in pastures for driven livestock breeding, stock driving

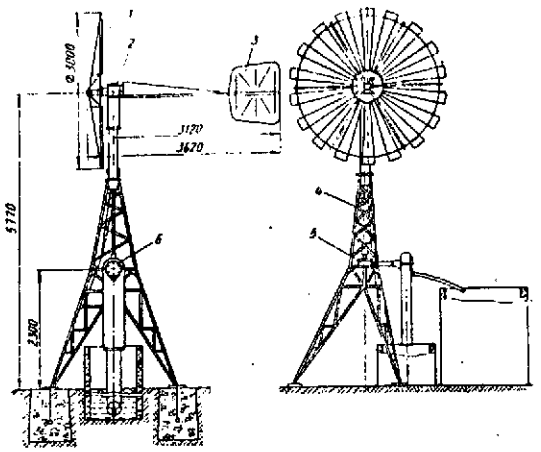

Fig. 47. Overall view of VTL-3 machine: 1. Windmill; 2. Head; 3. Tail; 4. Tower; 5. Lower reduction gear; 6. Water lift.

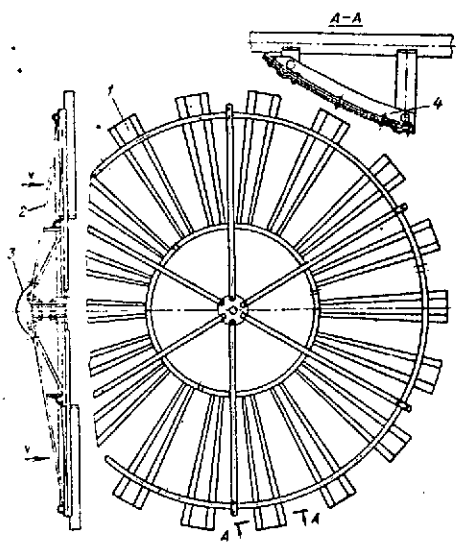

Fig. 48. Windmill of VTL-3 machine: 1. Blade; 2. Frame; 3. Cap; 4. Positioning angle bracket.

trails, livestock breeding farms and field mills, in small settlements, installing it directly over the water source. The average annual wind velocity in the area must be higher than 3.5 m/sec.

The machine consists of windmill 1 (Fig. 50), head 2, tail 3, three-edged tower 4, water lift (pump) 5 and manual control mechanism 6 with a device for automatic starting and stopping the machine.

The windmill is the high-speed type, with two duraluminum blades fitted with a system for centrifugal-aerodynamic adjustment of rotation frequency. The stems of the blades are fastened in ball bearings in the hub. Therefore, the blade, depending on the rotation frequency of the windmill and the wind velocity, turning due to the action of the regulator, /86 can be set at the proper angles of inclination ϕ: the higher the velocity the lower angle ϕ.

The head consists of a cast body 1 (Fig. 51), upper section 4 of the tower with pivot pin 3 and two support bearings 2, which serve for free rotation of the head during orientation relative to the wind direction. On the end of windmill shaft 8 is a crankshaft, on which rod 7 is installed in two bearings. Swivel 6, fastened on the lower end of the rod, permits rotation of the head relative to its vertical axis.

Using a conic pin, the windmill hub is fastened to the main shaft, which turns in two radial ball bearings located in the head body.

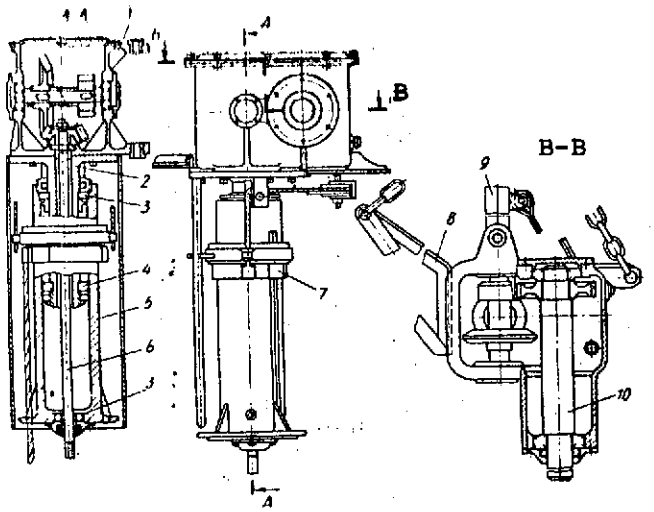

Fig. 49. Head of VTL-3 machine: 1. Housing; 2. Support pipe; 3,4. Radial-support bearings; 5. Guide; 6. Vertical shaft; 7. Swivel sleeve; 8. Regulator bracket; 9. Tailpipe; 10. Main shaft.

In the lower part of the head beneath the swivel is installed central spring 5, whose compression force compensates the weight of the water lift pipe, easing starting of the machine and making its operation smoother. The tail, consisting of angle brackets and an empennage is fastened to ribs on the head.

The water lift consists of separate sections of pipe, which are connected together using flanges 1 (Fig. 52), bushings 3, and bolts 2. Water is fed from the cast transfer sleeve 4, installed after the upper section of tubing into the pressure pipe or drains into the tank. Body 8 of rubber valve 6 is fastened onto the lower section of pipe using sleeve 9 on threads. The height of its lift is 3 mm. /87

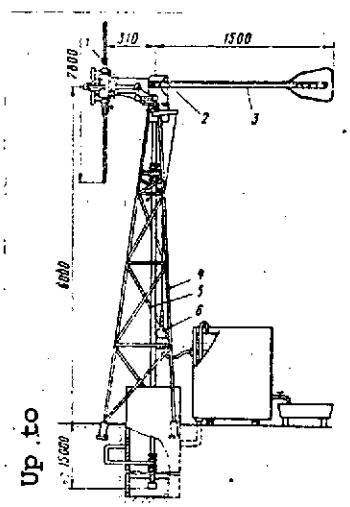

Fig. 50. VB-3T machine.

To decrease lateral oscillation, the pipe is centered along the tower with flexible tensioners made of rubberized belt, located in three tiers.

The water lift works in the following manner. When the windmill rotates, the crankshaft drives the water lifting pipe with the pump in a back-and-forth motion with a frequency of up to 420 oscillation per minute and a stroke (amplitude) of 46 nm. When the pipe moves downward, the valve lifts and admits a portion of water into the pipe (position I). At the beginning of the pipe's stroke upward, the valve closes the passage holes, and the column of water, together with the tube, rises upward with some speed (position II). During the next

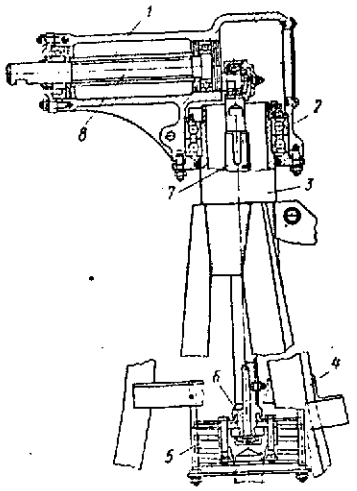

Fig. 51. Head of VB-3T machine.

stroke of the tube downward, the water column, due to inertia, continues to move upward, forming a suction above the valve. As a result of this, the valve again lifts and admits water from the water source into the pipe (position III). As the result of the rapid movement of the pipe upward and downward, the water is pumped into the tank or the water pressure tower. When the tank is filled with water, the machine is stopped as the result of engagement of the automatic device (see Chapter 12).

The regulation mechanism of the rotation frequency of the windmill consists of two centrally located springs 10 and 11 (Fig. 53), centrifugal weight 3 and parts for kinematic connection of the blades: guide bushings 9, sliding along a shaft; adjustable tension nut 13 and two carriers 12. /88

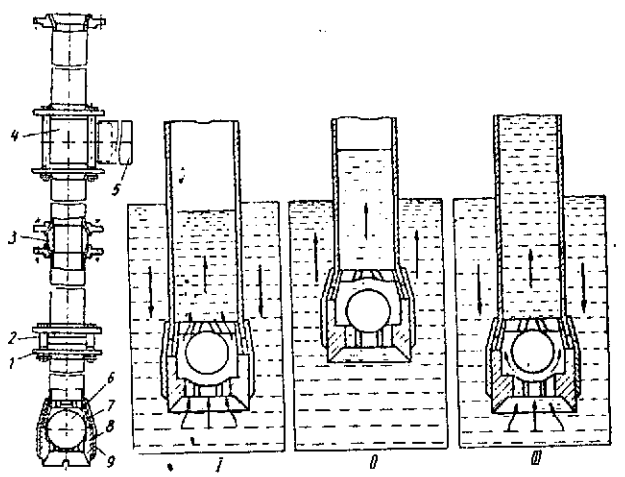

Fig. 52. Principle of operation of the inertia water pump: I. downward stroke of pipe -- filling of water pump pipe with liquid; II. upward stroke of pipe -- feed of liquid; III. downward stroke of pipe -- new cycle of filling pipe with liquid; 1. Flange; 2. Bolt; 3. Bushing; 4. Transfer sleeve; 5. Rubber hose; 6. Rubber valve; 7. Lift limiter of valve; 8. Valve body; 9. Sleeve.

The weights are connected with the blades, and through the carriers and nuts, also with sleeve 7 of the regulator. When the starting (small) spring 10 is compressed, the sleeve slides along bushing 9, and when the mainspring 11 is compressed, the sleeve, together with the bushing moves along the regulator rod.

If the windmill is stationary, sleeve 7 is pressed against nut 8 on the end of regulator rod by the springs and the blades occupy the starting position I (Fig. 53), i.e. they are set at a large angle. In this case, the machine begins to work at a low wind velocity. As the frequency of rotation of the wheel increases, the centrifugal forces of the weights, overcoming the

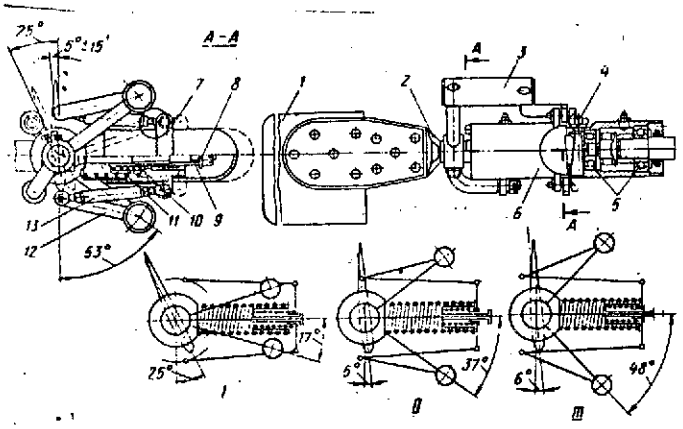

Fig. 53. Construction and position of windmill adjusting mechanism of the VB-3T machine: I. starting; II. working; III. during adjustment; 1. blade; 2. neck; 3. regulator weight; 4. wedge; 5. bearings; 6. bushings; 7. sleeve; 8. support nut; 9. guide bushing; 10. starting spring; 11. mainspring; 12. carrier; 13. adjustable tension nut.

resistance of the small spring, move the regulator sleeve and rotate the blades from the starting position into the working one II.

With a further increase in rotation frequency, the centrifugal forces of the weights compress the mainspring, turning the blade in a direction toward decreasing the angles of inclination. With this, the aerodynamic rotation moment of the windmill decreases and the frequency of rotation is decreased.

At the moment regulation begins, the angles of inclination become negative (Fig. 53, position III). With a reduction in the velocity of the flow and a decrease in rotation frequency of the windmill, the centrifugal forces also decrease, and the mainspring turns the blades into their initial working position. When the windmill, as the result of a reduction in wind velocity, stops, the small spring turns the blades into the starting position. The regulator maintains the nominal number of revolutions with a deviation within limits of 10-15%. /89

The TVM-3 "Chayka" machine is designed for lifting water with a piston pump from pit and pipe wells having a depth of up to 30 m in regions where average wind velocities are 3-5 m/sec. With a NP-65 pump, the machine can be used for lifting water from a depth of up to 50 m, although with this its productivity is decreased. Installation with this machine is recommended for summer pastures, in settlements and on small livestock breeding farms. /90

The wind motor of the machine has a multibladed windmill and begins operation at wind velocities of 3-3.5 m/sec. The motor is mounted over the well.

The machine consists of the following components: windmill 1 (Fig. 54), tail 2, head 3, mast 4, frame 5 with manual drive, pumps 6 and manual starting and stopping mechanism 7. The set also includes hoses, water lifting pipes 8 and a lift boom.

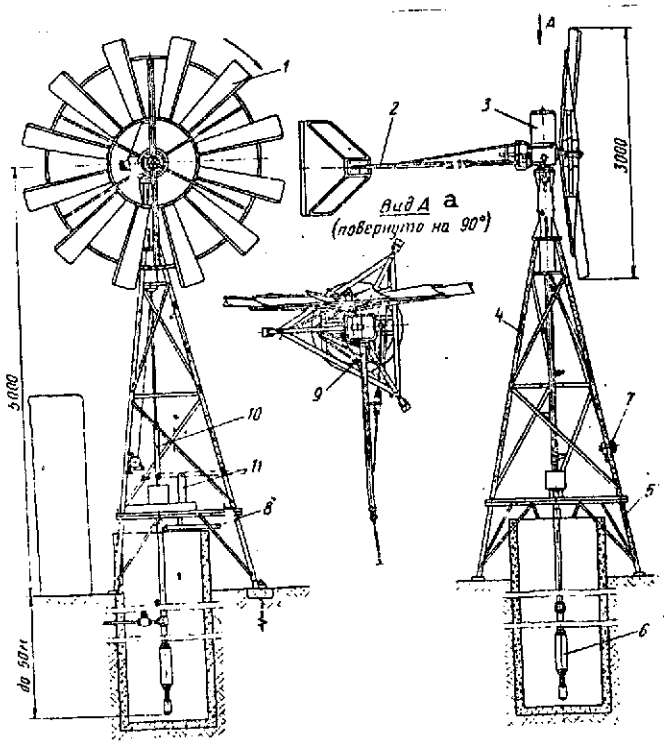

Fig. 54. "Chayka" machine.

Key: a. View A (turned by 90°)

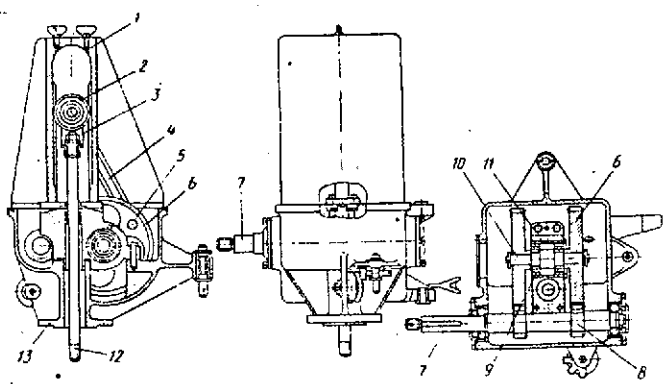

Fig. 55. Head of the "Chayka" machine.

The windmill has 12 steel blades, fastened on a frame of angle irons. The leading edges of the blades are crimped for higher strength. The windmill hub is fastened onto the shaft of the head.

The head is a single-stage reduction gear with two pairs of straight gears 6 and 8 (Fig. 55) and a crankshaft-connecting rod mechanism, located in an aluminum body 13. The small gears 8 are made as a unit with main shaft 7 and, from it, drive gear 6 with pins 5, on which connecting rods 4 are mounted. The heads of the connecting rods are connected with axle 10 and drive rod 12, which goes to the pump rod in a back-and-forth motion through traverse 3. Bearing 2, which is located along guide 1 is fitted onto shaft 10.

The shaft of the windmill is displaced relative to the axis of rotation of the head. The frequency of rotation and power of the windmill are limited by its output due to the wind. The head is connected with the support pipe, which is fastened to the mast, through the support-radial bearing and the sliding bushing.

The mast, an open three-edged frame of pipe and angle steel, /91 has unified construction. The rod passes inside the mast, and on one of the legs is a toothed starting and stopping windlass. Catch-lock 9 for storm protection (see Fig. 54) is fastened onto

pipe of the tail, and at wind velocities higher than 17 m/sec, fixes the tail and head into the "stopped" position. This eliminates harmful "flapping" of the windmill during strong wind gusts, decreasing gyroscopic loads and increasing the reliability of the machine. The catch-lock is disengaged using the windlass, slightly pulling on the cable.

The lower frame is installed above the well, fastened to the ground with anchor bolts and connected with the tower with hinges. The transfer box with pipes and the pump, having two valves and an inlet filter screen is fastened to the frame. The walls of the pump cylinder are chromed, and the cuffs are made of special rubber. The piston is driven through thin-walled sealed rods, which eases operation of the motor.

For using manual drive, insert 10 of the rod (see Fig. 54), fastened onto support axis 11, is tilted away.

The 2VPL-4 "Buran" machine is designed for lifting water into a tank at a height of up to 2 m from pit wells having a depth of up to 30 m, primarily in regions of driven livestock, where the average annual wind velocity is equal to 4-6 m/sec. It is used for lifting water in cases where feeding it to a great height above the surface of the earth is not required. Storage batteries can be charged simultaneously from the motor, for which it is fitted with a G-2B alternating current generator with a power of 700 W and 6-ST-128 batteries. The machine is installed as a distance of 2-2.5 m from the water source. The machine consists of windmill 1 (Fig. 56), upper reduction gear 2, tail 3, side plane 4, tower 5, lower reduction gear 6, belt water lift 9, starting and stopping mechanism 10, a frame for fastening the water lift and belt drive guard 8, generator 7 and an electrical control panel.

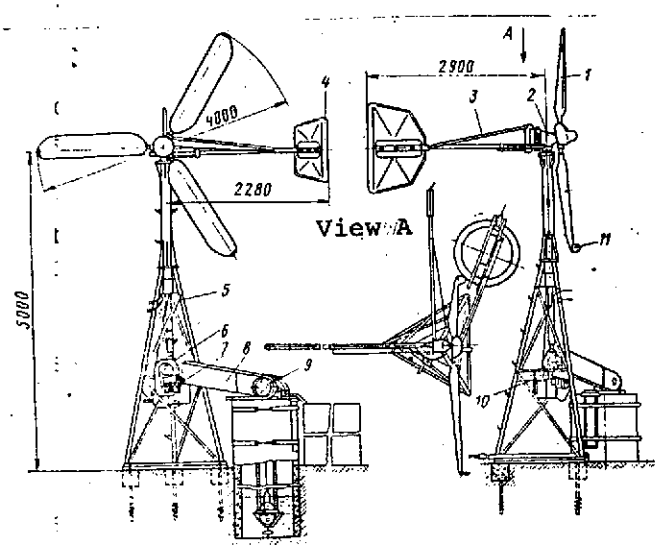

Fig. 56. "Buran" machine.

The windmill has three fiberglass blades, fastened onto a hub, covered by a housing. For more reliable operation of the regulation system, tips 11 are fastened onto the ends of the blades. The blades are slanted from the plane of rotation of the windmill by 5°.

The upper reduction gear (Fig. 57) is a pair of conic gears (i = 1 : 2.18) located in bearings with one-time lubrication. The reduction gear has an aluminum cast body. The reduction gear is fastened

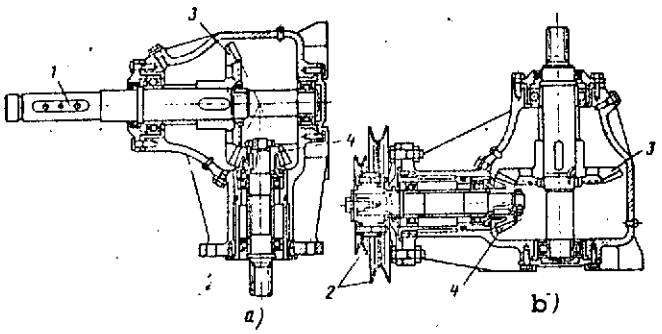

Fig. 57. The unified reduction gears of the "Buran" machine: a. upper; b. lower; 1. main shaft; 2. drive pulleys of water pump and generator; 3 and 4. gears.

by its lower flange to the rotating support, which has a support-radial bearing with free balls; a bracket for fastening the bearings of the tail and the side plane tube is connected to the lateral flange of the reduction gear with bolts.

The power of the machine is limited automatically by turning the windmill out of the influence of the wind, using the side plane. The machine is started and stopped in the same way as the "Chayka" machine. The catch-lock devices in them are also identical. /92

The pipe tower has bearings in its lower part. It is installed on a base made of angle steel, which is fastened to the earth with three earth augers.

The lower reduction gear (see Fig. 57) is interchangeable with the upper one and is installed in the same way as the generator, on an intermediate diaphragm in the tower. Two pulleys are mounted on the horizontal shaft of the reduction gear: one for driving the V-belt of the water lift, and the other for the generator.

The water lift has a perforated rubberized belt with dimensions of 60 x 4 mm. Its body is fastened to the head pipe of the well with clamps. If necessary, the water lift can be raised to a height of up to 2 m from the earth.

The generator charges storage batteries with a wind velocity of 6 m/sec and higher, when the frequency of rotation of the shaft reaches 900 rpm.

The electrical diagram of the machine (Fig. 58) consists of a type G-2B synchronized generator SG (P = 750 W; U_n = 24 V; n_n = 1500 rpm), with a separating excitation winding WI, rectifiers B_1 and B_2, a type RR_5 relay- regulator and a storage battery batt. At a certain rotation frequency, the generator, due to its residual electromotive force, fed to rectifier B_1, kicks in the relay regulator RR_5, voltage is fed to the excitation winding WI and self-excitation of the generator occurs. During the process of operation, the generator, through rectifier B_2, charges the storage battery, from which the load is powered. /93

Control over the operation and charging of storage batteries is accomplished using an ammeter and voltmeter. At high wind velocities, the voltage is limited by relay-regulator RR_5.

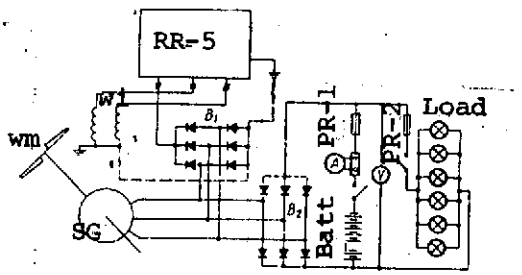

Fig. 58. Electrical diagram of the "Buran" machine.

The work of the motor with the water lift and generator is possible with wind velocities higher than 7 m/sec.

The VNP-4M "Vikhr'" pneumatic pumping wind machine is designed for lifting water from wells up to 30 m deep and from open water reservoirs. It can be used in regions with an average annual wind velocity over 4 m/sec and in areas with frequent storms. The machine is installed to the side of the water source at a distance of up to 100 m and is combined with a reserve tank having a volume of not less than 20 m^3.

Some components of the machine (the support device, tower, orientation and starting-stopping mechanisms) are interchangeable or similar in installation with components of other machines (VPL-4, VEN-4). According to the wishes of the user, the "Vikhr'" wind machine is assembled with a pump for lifting water from bored wells with a diameter of 100 mm and greater. /94

The machine consists of the following basic components: windmill 1 (Fig. 59) with a centrifugal rotation frequency regulator, diaphragm compressor 2 with a device for automatic unloading of the windmill during the starting and acceleration period, turning support 4 with a system for manual starting and stopping of the motor and a worm wheel device for automatic orientation of the head during changes in the direction of the wind, two wind roses 3, upper support-receiver 5, lower mast 6, mast base 10, and pneumatic pump 12 with an automatic distributor device 11. /95

The windmill with two blades made of pressed fiberglass is fastened to the shaft of the compressor using a wedge and is located behind the mast. The blade, using its tip, is fastened to its root with six bolts, and the root turns in the hub-bushing 2 on two ball bearings 3 (Fig. 60), as the result of which it can rotate relative to the longitudinal axis during regulation.

The automatic system for centrifugal-aerodynamic regulation acts in the same way as in the VBL-3 and VB-3T machines. The regulator provides normal operation of the machine throughout the entire range of working wind velocities up to 40 m/sec. Regulation is accomplished by rotating the blades in the direction

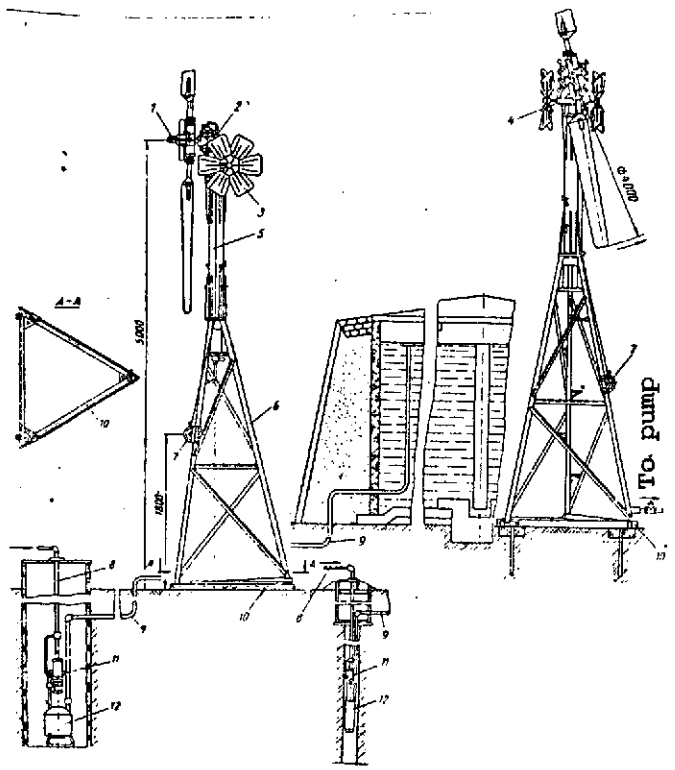

Fig. 59. "Vikhr"" machine: 1. Windmill; 2. Compressor; 3. Wind rose; 4. Turning support; 5. Support-receiver; 6. Lower mast; 7. Starting and stopping handle; 8. Air line; 9. Water-lifting pipe; 10. Mast base; 11. Pump automatic device; 12. Pneumatic pump.

"towards the flow," i.e. in the direction toward decreasing the angles of inclination. Using the regulator, the machine can be started into operation automatically with optimal angles of inclination of the blades (30-31°). The blades are set at a high angle using starting spring 8. Working spring 7 is extended beforehand and begins to operate with a windmill frequency of rotation of about 290 rpm.

The system for manual starting and stopping is connected to the regulator. The system consists of rod 2 (Fig. 61), which passes through the hollow shaft of the compressor. One end of the rod is installed in bearings in housing 9 (see Fig. 60), and the other is connected through lug 3 (Fig. 61) with the rod of the relieving device 4 of compressor 1. Lever 6 is fastened onto the upper plate of turning support 7 on a pin. Its short end /96 is connected through two connecting rods to slide 8, where the return spring is compressed downward toward the rollers of the two-armed lever 13. Using cable 11, the lever is connected with the starting and stopping handle. The blades are fixed in three positions by the lever: "to start" (ϕ = +30-+31°), "to stop" (ϕ = -7°) and "at rest" (ϕ = +1°).

The compressor has four working chambers, each of which is formed by cover 15 with suction valve 17 and pressure valve 18 /97 and an elastic diaphragm made of rubber-impregnated fabric. The diaphragms are set into operation through pistons which are connected with two push rods connected in opposition. The push rods, moving back and forth in graphite bushings, are set into motion by eccentric shaft 14, which rotates in two bearings, which are located in cavities in the aluminum housing. Atmospheric air is sucked in through filters 16 and pumped into the collector manifold 19. The later is connected to the tube-receiver 9, and with the body of the by-pass (relieving) device 4.

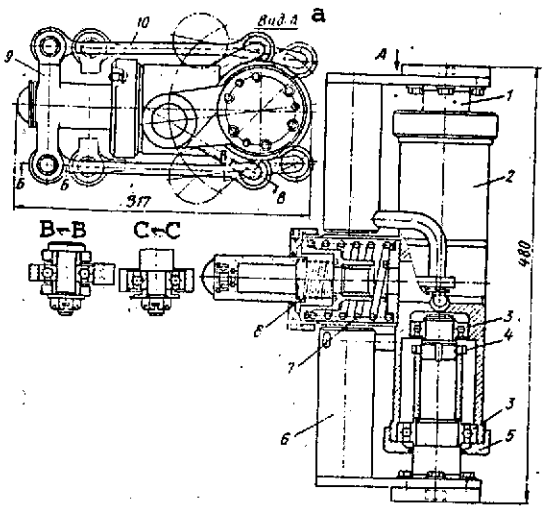

Fig. 60. Windmill and regulator of "Vikhr'" machine:
1. Blade root; 2. Hub; 3. Bearing; 4,5. Nuts; 6. Weight with linkage; 7. Working spring; 8. Starting spring; 9. Kinematic connection housing; 10. Connecting rod.

Key: a. View A

When the windmill is stationary or turning slowly, when the starting spring is not totally compressed, the plunger of the by-pass device, having two sealing rings, is located on the bottom: the by-pass holes are open. Air from the compressor passes into the atmosphere, counterpressure is not created, and the compressor works at idle. This eases acceleration of the windmill. When the windmill reaches 100-120 rpm, the regulator compresses the starting spring, and the plunger is moved and covers the holes. Compressed air begins to flow into the tower-receiver. A check valve is installed on the pressure mainline and prevents air from escaping from the tower when the windmill is stopped.

The turning support, together with the compressor 1 and wind roses 5 (see Fig. 61) makes up the head. It includes intermediate column 7 (Fig. 62), which is fastened to the mast, the mechanism for automatic orientation, the support-radial bearing 2 with boured balls 3 and supporting wire rings 9, /98 pressure pipe 5, seal 6 and lever 1 of the starting and stopping mechanism. Clearances in the bearings are adjusted with washers 10.

The worm wheel 8 of the orientation mechanism is fastened stationary onto the column and worm 4 with the wind roses rotates in bearings installed in cavities in the support. When the direction of the wind changes, the wind roses, which are located in front of the windmill for greater sensitivity, begin to turn, and, with the worm mechanism, the head rotates until the plane of rotation of the windmill becomes perpendicular to the direction of the flow. The intermediate column is connected to the upper support with bolts.

The upper support-receiver is a sealed tube with safety valve 10 (see Fig. 61), a pipe with a hole for drainage of condensation and a nipple for connecting it with the lower mast. For better hermetization of the tower, there is a sealing device 12.

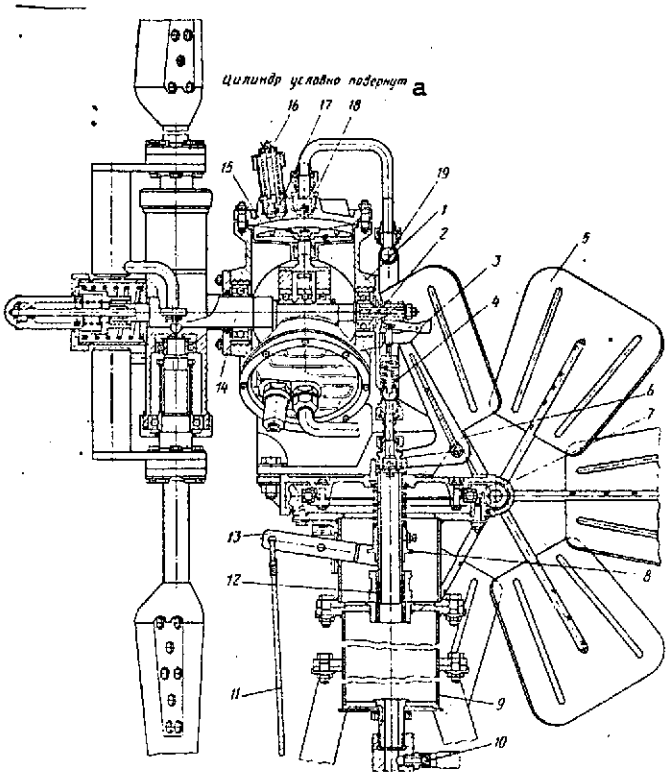

Fig. 61. Head of the "Vikhr'" machine: 1. Compressor; 2. Regulator rod; 3. Lug; 4. Relieving devide; 5. Wind rose; 6, 13. Levers; 7. Turning support; 8. Slide; 9. Tube-receiver; 10. Safety valve; 11. Cable; 12. Sealing device; 14. Eccentric shaft; 15. Cover; 16. Filter; 17. Suction valve; 18. Pressure valve; 19. Manifold.

Key: a. Cylinder abitrarily turned

The mast, of unified construction, has three tubular uprights, which are connected by buides, crossties, struts and a bulkhead. One of the uprights is used as the air pipe to the pump.

The water lift is a single-chambered pneumatic pump, consisting of a body 1 (Fig. 63) with suction valve 4, float 6 with rod 5, pressure pipe 2 with check valve 3, air pipe 8 and automatic distributor device 10 with double-seat valve 9.

Due to hydrostatic pressure, water raises valve 4 and fills the chamber of the pump. With this, the float is raised upward to ring 7 on rod 5 and moves further together with the rod. When the float is in the upper position, the distributor device 10 transmits an impulse to the two-seated valve, due to which air from the tower-receiver flows into the pump and begins to force water out along pipe 2. When the pump is totally evacuated, the float occupies the extreme lower position, and the air which remained in the chamber after the water was forced out, passes out through the two-seated valve into the atmosphere, pressure in the pump is reduced to normal and the chamber is filled with the next charge of water. /99 The working cycle is then repeated.

The machine works in the following manner. With wind velocities of up to 3.5 m/sec, the windmill, with the blades turned to the starting angle, stands motionless; the pressure chamber of the compressor is connected to the atmosphere through the by-pass device, and, therefore, the machine begins to operate when the compressor is still not loaded. With an increase in wind velocity, the windmill begins to turn. As the frequency of

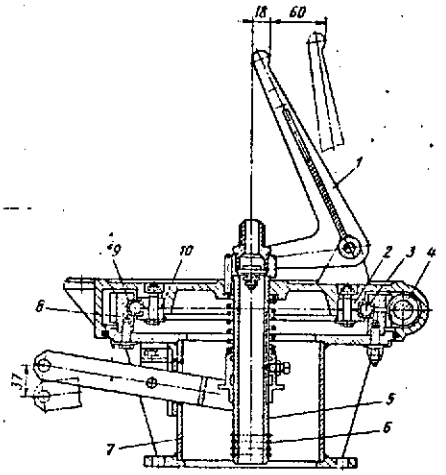

Fig. 62. Support-turning device of the head of the "Vikhr'" machine.

rotation of the windmill increases, the weights, due to the effect of centrifugal forces gradually compress the starting spring. The regulator rod moves and, when the small spring is totally compressed, the holes of the by-pass device are covered over and compressed air begins to flow into the tower. With an increase in wind velocity higher than calculated, when the rotational moment of the windmill is greater than the moment of resistance of the weights, they, due to the effect of centrifugal force, compress the mainspring of the regulator, the blades turn to smaller angles, the rotational moment is decreased and the frequency of rotation is limited to 290-320 rpm. With a reduction in the wind velocity and stoppage of the windmill, the regulator moves the blades into the starting position, simultaneously opening the by-pass device. Pressure in the cylinders falls to zero, and therefore the compressor is unloaded by the beginning of a new start. Due to the check valve on the pressure mainline, pressure in the receiver is sustained, as the result of which the consumption of compressed air is decreased. When the machine is stopped by force it is done manually. /1(

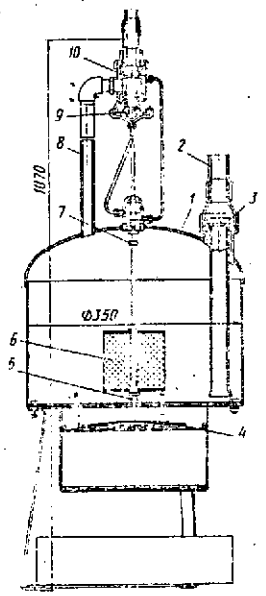

Fig. 63. Pneumatic pump.

The VEN-4 "Berkut" electrical pumping machine is designed for pumping water in regions characterized by an average annual wind velocity of 4 m/sec and higher. It can be used on low-yield pit wells with a limited stratum of water (up to 0.7 m) and pipe wells with casing pipes 100 mm and greater in diameter. For charging storage batteries, illumination and establishment of radio communications at shepherd brigade sites and field camps, a special charging device is fitted to the machine.

The machine consists of the following components: windmill 1 (Fig. 64), reduction gear 2, generator 3, turning support 4 with a wind rose orientation mechanism, tower 5 with support frame 7, electrical panel 6 of the automation and protection system for the machine, a submerged 9 or floating 11 electric pump, water lift pipe 8, power cable 10, and starting and stopping handle 12.

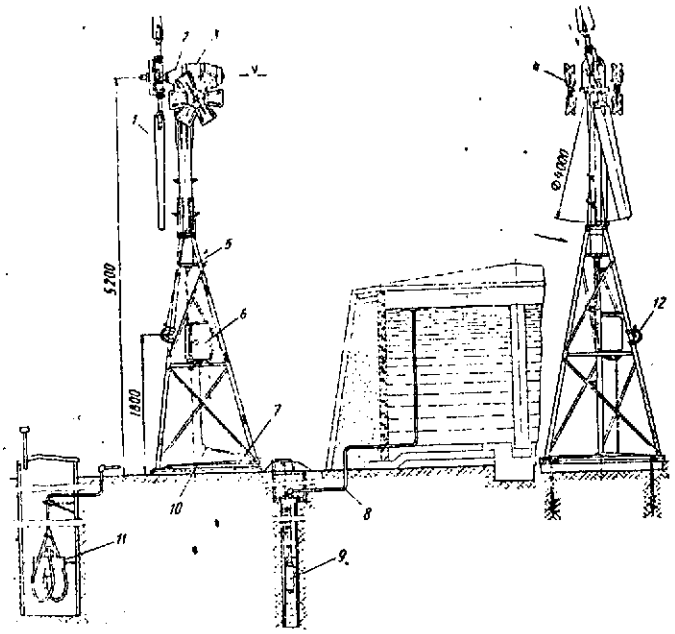

Fig. 64. "Berkut" machine.

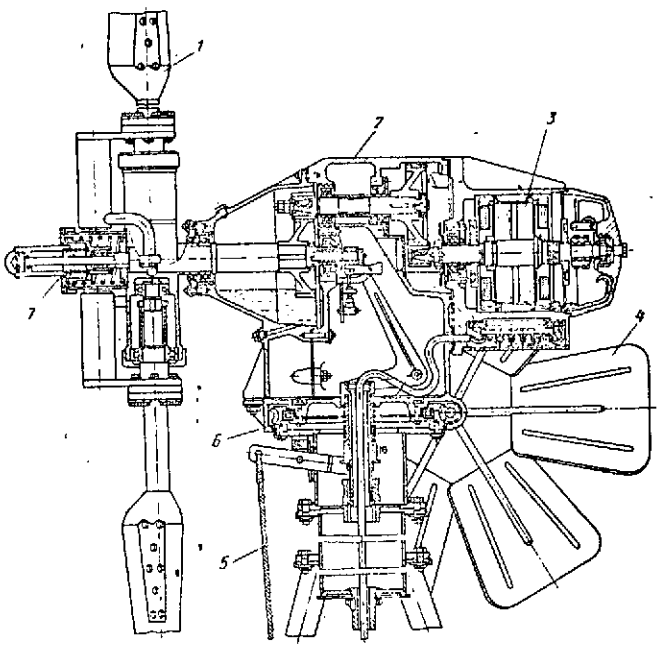

Fig. 65. Head of the "Berkut" machine: 1. Blade; 2. Reduction gear; 3. Generator; 4. Wind rose; 5. Starting-stopping cable; 6. Orientation mechanism; 7. Regulator.

The windmill, reduction gear, generator and turning support with the wind rose mechanism for orienting the windmill to the wind form the head of the machine (Fig. 65), after assembly. The windmill, reduction gear, turning support, orientation mechanism and tower are the same as in the "Vikhr'" machine.

The generator, type GVEN-T/230-ch/150 is three-phase with self-excitation from solid rectifiers. The cable running from the generator is fastened to the plate of the turning support and passes through a central hole in the guide pipe to the electrical panel. The generator is fitted with a closed transfer flange and housing, which encloses the rear bearing.

The electrical panel provides self-excitation of the generator at idle with a constantly switched-on pump and visual observation of the work of the machine, regulates the voltage according to a set rate, determines the best operation of the machine with a changing frequency of rotation and protects the machine from overloads and short circuits. The panel is included in a body, on the cover of which are instruments, automatic switching buttons and a panel of cable connectors for connecting the generator, electric pump and charging device.

The electrical diagram of the panel is presented in Fig. 66. Self-excitation of the generator at idle and with the pump turned on is provided by a residual magnetic field and a three-phase rectifier made of semiconductor diodes. Stable self-excitation of the generator is achieved with frequencies of 90-165 Hz, both during idle and during the operation of the pump, and the voltage is changed according to the law $u/f \approx const$, where u is the generator voltage; f is the frequency of the current. /101

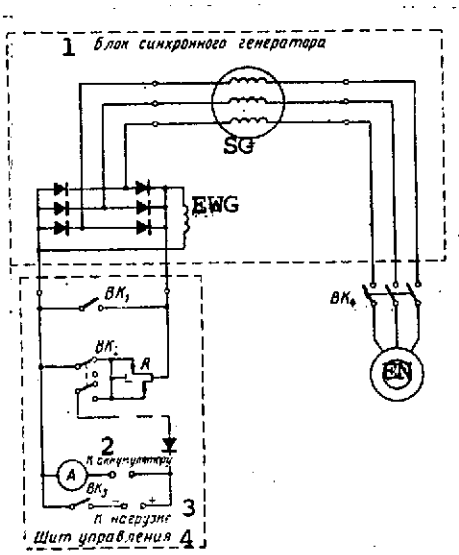

Fig. 66. Electrical diagram of the "Berkut" machine: SG - synchronized generator; EWG - excitation windings of generator; BK_1 - generator switch; BK_2 - load transfer switch; BK_3 - load disengaging switch; BK_4 - AP-50 automatic switch; EN - asynchronized pump drive motor.

Key: 1. Synchronized generator block; 2. To storage battery; 3. To load; 4. Control panel

The electric pump has a motor, on the cantilevered shaft of which is mounted the working wheel. The motor and working components of the pump are located in a body. Water pumped by the pump passes between the bodies of the motor and the pump, and flows into the pressure pipe through holes in the bearing body.

A support plate with a measuring insert, designed for adjusting the operating rate of the pump depending on the depth of water level in the well and the water source yield, serve for fastening the pump in the well.

The VNV-4 "Veterok" high-speed wind machine with single-screw pump (Fig. 67, a) is designed for lifting water from a depth of up to 30 m from pit and pipe wells and can be used in all zones where the average annual velocity of the wind is greater than 3 m/sec.

The basic components of the machine are (Fig. 67, b): windmill 1 with regulator, upper reduction gear 2, turning support 3 with wind rose orientation mechanism 4, upper pipe 5 and lower mast section 6, vertical shaft 7, manual drive 8, pump 9 with pipes 10 and manual starting and stopping mechanism 11. /102

With the exception of the windmill, the reduction gear and manual drive, the remaining components are similar to components used in the "Vikhr'," "Berkut" and "Buran" machines.

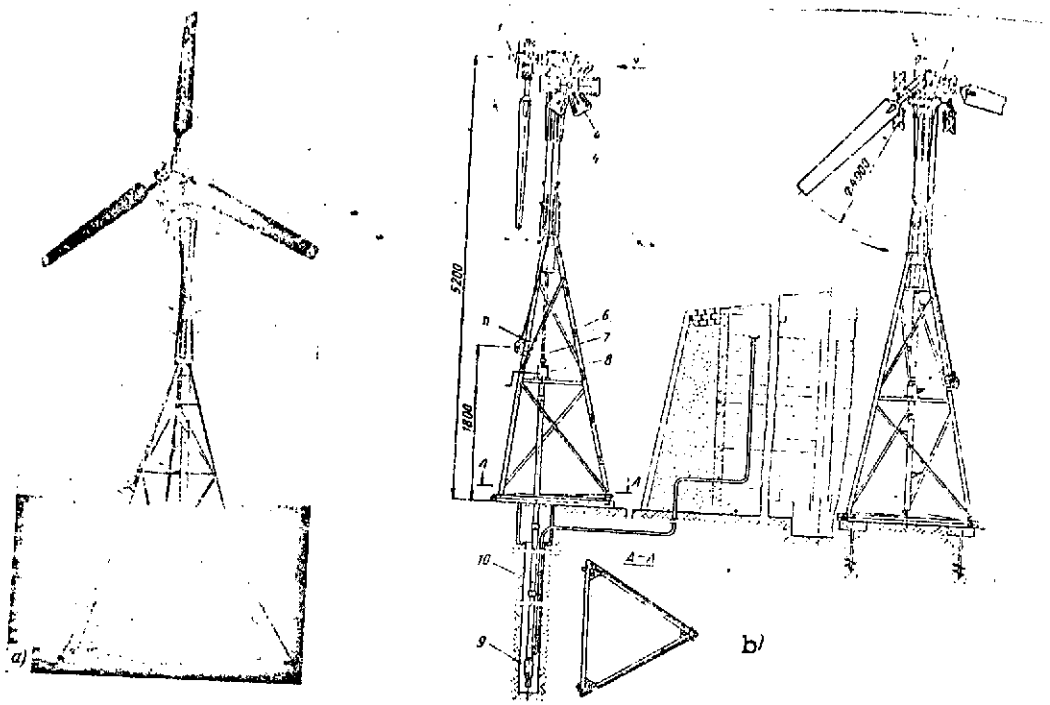

Fig. 67. "Veterok" machine.

The windmill is located behind the tower and has three fiberglass blades. As in machines with two-bladed windmills, the blades, during adjustment, are rotated by the effects of centrifugal and aerodynamic forces in the direction towards a decrease in the angles of inclination. Optimized starting of the machine at an angle of ϕ = 30-31° is used.

The windmill hub is fastened onto the shaft of the single-stage conic reduction gear (1 - 1.3), together with other components forming the head (Fig. 68). Rod 3 of the manual control passes through hollow shaft 1 of the reduction gear and, when the machine is stopped, is drawn out to the right by lever 5. The body of the reduction gear is covered on the side of the wind flow by housing 4. The rotational moment of the windmill is transmitted to vertical shaft 6, which, using a dog clutch, is connected with the manual drive and the pump shaft.

During operation of the motor, the drive handle is withdrawn from engagement and remains stationary. Using the manual drive, the dog clutch moves upward and fastens onto the vertical shaft. /105

The manual drive is a single-stage conic reduction gear with a gear ratio from the handle to the pump shaft equal to 1 : 2.7.

The pump is driven into rotation by the vertical shaft, consisting of sections 2 m each in length and centered relative to the water lifting pipe by rubber guide bearings.

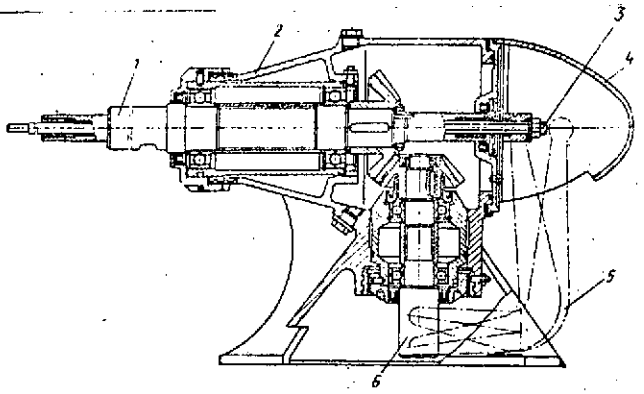

Fig. 68. Head of the "Veterok" machine: 1. Main shaft; 2. Body; 3. Rod; 4. Housing; 5. Lever; 6. Vertical shaft.

The tower can be made in two variations: normal and inclined (see Fig. 136, a). The utilization of an incline tower provides the possibility for installing the motor next to the water source without loading the well head. With this, assembly and servicing of the installation are significantly eased.

The productivity of water-lifting and pumping machines is presented in Table 15.

TABLE 15. PRODUCTIVITY IN m^3/h OF LOW-POWER WIND PUMPING MACHINES

Type of machine	(a) Высота водоподъема в м	(b) Суммарный напор в м	Wind velocity in m/sec					
			3	4	5	6	7	8 и более (c)
VDL-3	30	30	—	0,1	0.35	1.1	2.3	3—3.3
VTL-3	30	30	—	0.25	0,9	1,5	2,3	2,7—2,9
VB-3T	10	12	0,6	1,8	3,0	3,1	3,1	3,1—3,2
	15	30	—	0,6	2,0	2,2	2,3	2,3—2,5
TVM-3 "Chayka"	8	10	0,55	1,0	1,35	1,7	2,0	2,3—2,5
	16	20	0,1	0,7	1,1	1,5	1,75	1,9—2,1
	25	30	—	0.6	1,0	1,4	1,7	1,8—2,0
2VPL-4 "Buran"	10	10	0,4	0.7	1.2	1,7	2,4	3,0—3,2
	20	20	—	0.5	1.0	1.5	2,1	2,8—3,0
	30	30	—	0.4	0.8	1.3	1,9	2,6—2,8
VNP-4M "Vikhr'"	8	10	0,4	1,8	2,2	2,6	2,7	2,7—2,8
	17	20	—	0,6	1,6	1,6	1,7	1,7—1,8
	26	30	—	0.5	0,7	0.7	0.75	0,75—0,9
VEN-4 "Berkut"	12	15	—	0,3	0,9	1,6	2,1	2,4—2,5
	21	25	—	—	0,6	1,2	1,6	1,9—3,1
VNV-4 "Veterok"	8	10	0,5	1,6	1,9	2,5	3,0	3,2—3,3
	16	20	0,3	1,2	1,5	1,9	2,3	2,8—3,0
	25	30	—	0.4	0,8	1,7	2,1	2,5—2,7

Key: (a) Height of water lift in m
(b) Total head in m
(c) ... and higher

§ 2. Electrical Wind Machines

Electrical wind machines are designed for eliminating shepherds' dwellings, field camps, deer herders' yurts, tents

and huts of various expeditions, rural clubs and also for providing power to types KRU-2 or KRU-10 radio installations in places where the average wind velocity is greater than 3.5 m/sec. Machines with a power of 0.15-1.5 kW are also used for cathode protection of major oil and gas pipelines, powering automatic meteorologic stations and distilling installations. Machines with a power of up to 1.5 kW operate with buffered low voltage (6-24 V) storage batteries.

The VE-2M machine has the following basic components: windmill 1 (Fig. 69), head 2, tail 3, stand 4, and an electrical panel with a storage battery.

The head of the machine is fastened, using the stand to support column 5. The column is fitted with guy wires 6, platform 7, spike steps 8, insulators 9 for suspending wires running from the generator to the electrical panel, and brake lever 10 for manual stoppage.

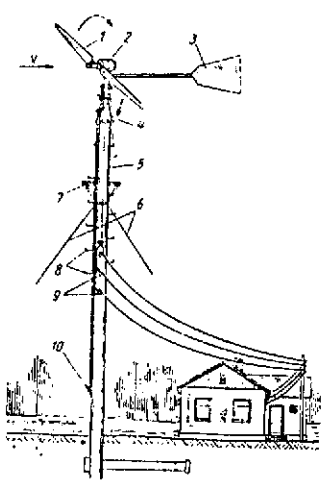

Fig. 69. VE-2M electrical wind machine: 1. windmill; 2. head; 3. tail; 4. stand; 5. support column; 6. guy wires; 7. platform; 8. spike steps; 9. insulators; 10. brake lever.

The windmill has metal blades (Fig. 70) whose roots are fitted into bearings in the hub. The hub is fastened onto the shaft of a generator.

Weights 3 are fastened on holders 2 to the root of the blade and they have pins 12, which are connected with sleeve 7 in an articulate manner through connecting rods 11. The sleeve slides along the shaft of the generator and rests on nut 13. The regulator of this machine works in the same way as the regulator for the "Berkut" machine.

On stand 3 (Fig. 71) carrying the tail and empennage, is fastened generator 1 with excitation from a constant magnet. In it is placed a three-phase stationary stator winding and a rotor in the form of an eight-pole constant magnet, which rotates in bearings which are mounted in the body, made of aluminum alloy. Depending on the method for connecting the windings, the generator puts out current with a voltage of 26 and 15 V. /106

The generator is connected with the electrical panel by a three-wire cable which passes through the stand pipe, which can rotate in a supporting ball bearing and guide bushing 4, which has a replaceable insert. The

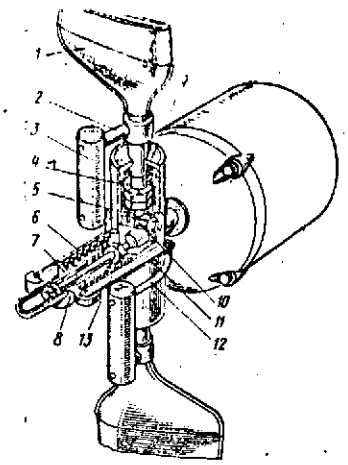

Fig. 70. Windmill and regulator of the VE-2M machine: 1. blade; 2. holder; 3. centrifugal regulator weight; 4, 5. ball bearings; 6. regulator spring; 7. regulator sleeve; 8. rod; 9. felt ring; 10. wedge with nut; 11. connecting rod; 12. crooked pin; 13. nut.

bushing is fastened onto the bracket 8 by which the machine is mounted on the support column.

The electrical panel has two selenium rectifiers, which are installed according to a three-phase double semi-periodic schematic, an ammeter 3 (Fig. 73) and voltmeter 2 for monitoring the operation of the machine, switches, fuses and clamps for connecting the load to the storage battery 5. In order to stop the machine, the shaft of generator 1 is braked. The cable pull of the brake is fed to a lever which is fastened onto the support column.

The machine can work with one or two types 6-ST-98 or 6-ST-128 storage batteries with a voltage of 12 or 24 V.

The VIESKh D-4 electrical charging wind machine is designed for providing power to agricultural radio communications installations, eliminating small livestock breeding farms, sites of field camps and shepherd crews and for charging storage batteries. /107

The machine consists of a windmill with a rotation frequency regulator, a reduction gear, generator, support with tail, support column with guy wires and lever for the starting and stopping mechanism, and an electrical panel. The windmill, with the reduction gear and generator, makes up the head (Fig. 73).

The windmill and rotation frequency regulator are the same as in the "Berkut" and "Vikhr'" machines.

The reduction gear is single-stage (i = 5.5). The large gear 13 is mounted on the main shaft 14 on a key and rotates in two covered ball bearings, which are fastened in body 7, made of aluminum alloy. The reduction gear is fastened to the tip of the support using stem 12, which has a tube and flange. An eye and cleats for fastening the guy wire of the tail is formed on the body of the reduction gear. The empennage is of trapezoidal shape, increasing the aerodynamic properties of the tail. The tip, which is designed for fastening the head to the support column, has support rings and bushings of textolite, forming supporting and radial bearings. The stem is fastened with lugs to the support column: wooden with side braces or reinforced concrete.

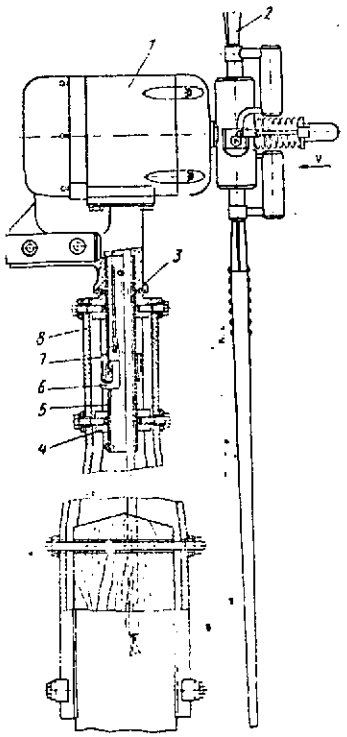

Fig. 71. Head of the VE-2M machine:
1. generator; 2. windmill; 3. stand; 4. bushing; 5. pipe; 6. hook; 7. sleeve; 8. bracket.

The generator, a type G-2B from a motorbus, has a power of 750 W.

The electrical panel is made in the form of an enclosed cabinet, in which are mounted a panel with instruments and apparatuses. The electrical schematic of the panel is similar to the schematic of a panel of a "Buran" machine. Powering of the voltage circuits of the RR-5 relays is accomplished from an auxiliary rectifier. The machine can work with a storage battery having a capacity of 128-250 amp h with a voltage of 12 V.

The UVEU-(1-4)-6 electrical wind machine is an independent source of power, designed for providing power to electric motors of pumps, small agricultural machines which are driven electrically, everyday appliances and for charging storage batteries. Depending on the wind conditions of the region, it is equipped with generators having a power of 1, 2 or 4 kW, with which the diameter of the windmill remains unchanged -- 6 m, and only the calculated velocity of the wind changes. The machine's set includes a ETsV-4-4-45 electric pump. At v = 7 m/sec and a head of 30 m, the productivity of the machine is up to 3.5 m^3/h. /109

Synchronized generator 3 is driven into rotation by windmill 1 (Fig. 74) with an automatic rotation frequency regulator through reduction gear 2. The head of the machine, besides the generator and reduction gear, has a worm reduction gear 4 for the automatic orientation mechanism and a wind rose 5. The head is fastened onto a tubular support (tower) 6 with a pivot hinge 12. The support has two stages of guy wires 8 and 13, and a lifting arm with winch 10. The cable for the starting and stopping mechanism and the electrical cable run inside the support. The set includes the automation block 7, pump 9 and foundations 11.

The windmill has two metal blades, which turn in the body in ball bearings. A radial-support bearing carries the load from centrifugal forces acting on the blade. For reduction of flyout of the windmill relative to the support, the blades are fastened in the body at an angle of 5° to the plane of its rotation toward the wind. The parts of the regulation mechanism are enclosed with a hood. /110

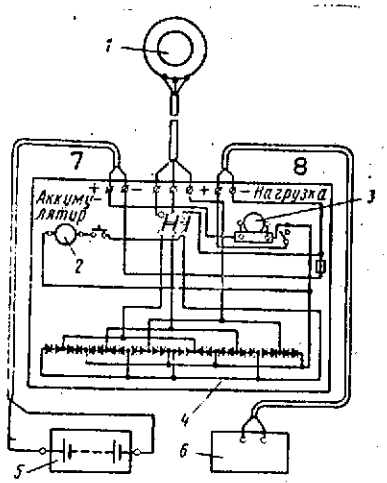

Fig. 72. Electrical diagram of the VE-2M machine: 1. generator; 2. voltmeter; 3. ammeter; 4. panel; 5. storage battery; 6. consumer; 7. to storage battery; 8. to load.

The regulator operates similarly to the regulators of the VE-2M machine, the "Berkut" and others. It has two springs: a starting and a working one, centrifugal weights, a kinematic connection rod and a limiter.

For forced stopping of the machine, tension on the cable is transmitted through a system of levers and linkage to the rod. The cable compresses the working spring, the blades are set in the non-working position and the machine stops. When the cable is released, the working and starting springs are freed and the blades take up the position "to start."

The reduction gear is two-staged (i = 8.06), and consists of two pairs of cylindrical gears located in a housing which is cast of aluminum alloy. The windmill hub is fastened onto the main shaft of the reduction gear. On the body is a flange for connecting the generator with bolts.

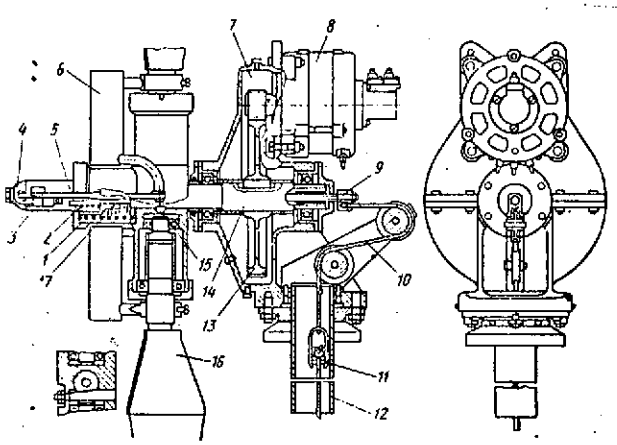

Fig. 73. Head of the VIESKh D-4 machine: 1. regulator working spring; 2. starting spring; 3. sleeve; 4. protector cup; 5. connecting rod; 6. regulator weight; 7. regulator body; 8. generator; 9. pull rod; 10. cable; 11. swivel; 12. stem; 13. large gear of reduction unit; 14. main shaft; 15. wedge-key; 16. windmill; 17. sleeve.

The generator is type SGV-1500, synchronized, with self-excitation from solid rectifiers, and is contact-free. The linear voltage with connection of the Δ/Δ phases is equal to 400/230 V. The shaft of the rotor rotates in two roller bearings. The stator, having a main and auxiliary winding, is located in an enclosed housing with the ribbed surface for better cooling of the windings. Fig. 75 shows the principal electrical schematic of the generator's self-excitation. /111

A system of automatic control of the

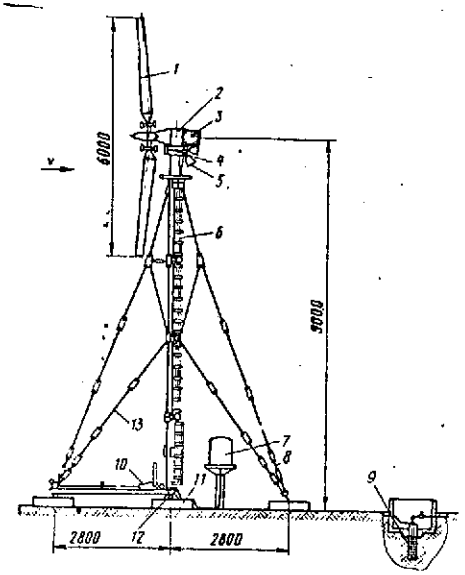

Fig. 74. UVEU-(1-4)-6 electrical wind machine.

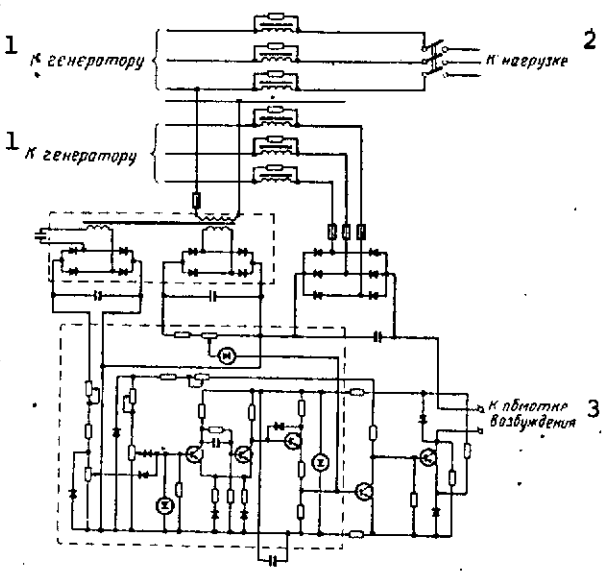

Fig. 75. Principle electrical schematic of self-excitation of the generator.

Key: 1. To generator; 2. To load; 3. To excitation windings

electric drive provides self-excitation of the generator upon attainment of a certain angular velocity by it; adjustment of the voltage according to frequency; limiting of the voltage at $\omega > 1.1\omega_{nom}$; removal of excitation upon a large decrease in rotation frequency or with an extreme increase in angular velocity of the rotor; repeated self-excitation of the generator at $\omega \approx (0.6\text{-}0.7)\omega_{nom}$.

The supporting device (Fig. 76) is a two-staged worm reduction gear with an overall gear ratio of $i = 2635$, which is installed on a flange in the upper section of the tower. The reduction gear of the head with the generator is fastened to its cast steel body. The weight of the head and the windmill is absorbed by bearings.

Worm wheel 3 of the automatic orientation mechanism is fitted onto the outer race of bearing 2, and the orientation mechanism has one six-bladed wind rose with a diameter of of 0.9 m. The wind rose is fastened onto the shaft of worm gear 10, which drives worm 8, which is engaged with wheel 3 through an auxiliary worm drive (not shown on Fig. 76).

The mast is a steel pipe consisting of five sections, connected with bolts on flanges. The starting and stopping winch is fastened onto the lower section.

Power developed by the machine is transmitted to the electrical panel along a three-wire cable.

The VEV-1-6 electrical wind machine is designed for providing power to small consumers (pumping and distilling installations) in regions not having centralized electic power. It is

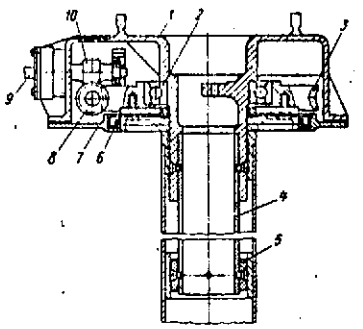

Fig. 76. Supporting device of the UVEU-(1-4)-6 machine: 1. body; 2. radial-support bearing; 3. worm wheel; 4. pipe; 5. bushing; 6. rubber cuff; 7. worm reduction gear cover; 8. auxiliary worm; 9. wind rose shaft; 10. main worm gear.

also used for charging storage batteries, illumination, providing power for radio equipment and other purposes.

The machine consists of a windmill 1 /112
(Fig. 77), reduction gear 2, synchronized three-phase generator 3, tower 6, and control and pumping station 7. The head includes a wind rose orientation mechanism 4, and a starting and stopping mechanism 5.

The windmill is two-bladed and has metal blades installed in the hub on three ball bearings. As in the "Berkut" machine, the centrifugal-aerodynamic system of adjusting the windmill's rotation frequency by turning the blades towards smaller setting angles is used.

An automatic device is used for starting the machine at the optimum starting setting angle of the blades. The regulator works the same as the regulator for the VE-2M and "Berkut" machines. The starting spring is compressed upon attainment of a frequency equal to 30% of the calculated rotation frequency of the windmill.

The nominal rotation frequency of the windmill (220 rpm) is achieved with a wind velocity of 6-7 m/sec. With wind velocity changing within limis of 4-25 m/sec, the rotation frequency is maintained within limits of 220-250 rpm.

The reduction gear is a step-up two-stage type (i = 13.56), and is the base of the machine's head. The windmill is fastened onto its main shaft, the generator is fastened to the body, and beneath the reduction gear is the orientation mechanism. The torque moment from the main shaft is transmitted to the output shaft, which is connected to the generator shaft through a sleeve.

The automatic orientation mechanism has a six-bladed wind rose 1 (Fig. 78) with a diameter of 1 m. The wind rose, through an intermediate worm pair 2 and 3 (i = 29) rotates the main worm gear 4, which can turn around worm wheel 5 (i = 81), which is fastened to the tower and set the windmill to the wind.

The tower is a three-edged conic truss with struts and four /113
bands.

The generator is synchronized, three-phase, enclosed, with natural blow-off by the airstream. There are two windings, a

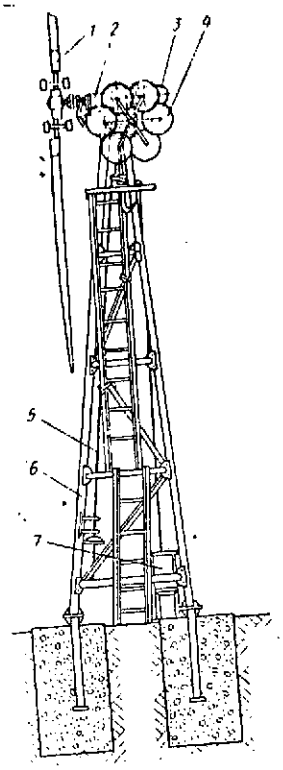

Fig. 77. VEV-1-6 machine:
1. windmill;
2. reduction gear;
3. generator;
4. automatic wind rose orientation mechanism; 5. manual starting and stopping mechanism cable; 6. tower;
7. control station.

power one and an additional one in the stator. The latter provides self-excitation of the generator. To improve self-excitation, the inductor is fitted with two permanent magnets, located beween the coils of the rotor poles. The generator shaft is connected with the output shaft of the reduction gear by a flexible sleeve.

The control station is made according to an electrical schematic (Fig. 79) which provides automatic excitation of the generator, switching on of the pump at working velocities of the wind, and also protection of the generator from overloads, and external and internal short circuits. The generator is excited by the manual button KV and automatically using the relay powered from the rectifier VBA.

When starting the machine, the load switches VN and automatics device VA are switched on. Until the windmill develops the required rotation frequency, the generator is not excited and current does not flow into the windings. When the velocity of the wind and rotation of the generator increase, it self-excites and sequentially kicks in relays 1RP and 2RP; the power winding is disengaged from the auxiliary, the load or the electric motor of the pump is switched on and the machine begins to provide water.

With a reduction in frequency down to 25 Hz, the generator loses its excitation and the pump stops. When exciting the generator using the KV button, switch VA in the automatics device must be turned off.

The "Sokol" electrical machine (2D-12MA) with a submerged centrifugal pump is designed for lifting water on farms, irrigated and reclaimed sections, and also for providing electrical power to small far-removed objects. The basic zones for the machine's use are Kazakhstan, Azerbaydzhan, near the Volga, the Arctic, the coastline areas of the Caspian, the Altay and others.

The distinguishing feature of the machine consists of the fact that, due to the use of a new system of moment-centrifugal regulation, it can be also used together with a non-wind electrical station for working in parallel with a network.

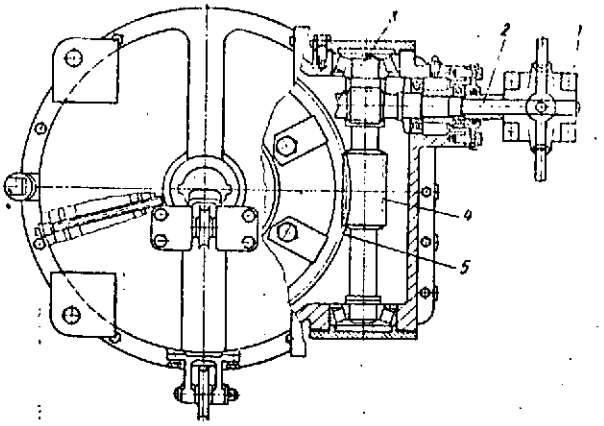

Fig. 78. Automatic orientation mechanism of the VEV-1-6 machine: 1. wind rose; 2. intermediate reduction gear worm; 3. worm wheel; 4. main worm; 5. worm wheel.

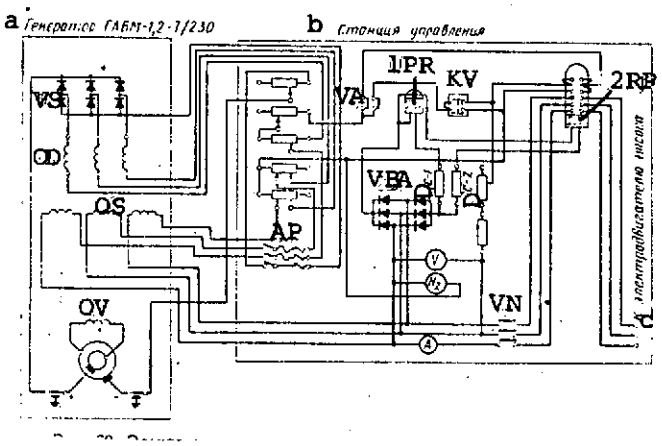

Fig. 79. Electrical schematic of control station of VEV-1-6 machine: 1RP and 2RP - relays; VC - rectifiers; DC - additional resistance; OD and OC - additional and power generator windings; OV - excitation winding.

Key: a. GABM-1.2-1/230 generator; b. Control station; c. To pump electric motor.

The machine has a three-bladed windmill 1 (Fig. 80), in the hub of which is built in a moment regulator 2; head 3, consisting of the reduction gear, generator and rotating support; orientation mechanism with two wind roses 4; pipe mast 5 with two levels of guys 9 and manual control winch 6. The equipment for the machine includes electrical panel 7 for control and automation, type EPN-6-16-110 pump 8, lifting boom and pile foundations.

The windmill is fastened onto main shaft 1 (Fig. 81) by the hub on bearings 2 so that a slight angular motion relative to the shaft is possible when the moving aerodynamic moment exceeds the calculated value. The fiberglass blades are fastened into the hub on bearings so that all the roots are kinematically connected together through rod 3 of the main shaft. To decrease flyout of the windmill relative to the tower axis, the shaft is installed at an angle of 10° to the horizontal. /115

The two-stage reduction gear 4 (i = 17.1) is connected to the rotor of the synchronized generator 5 on the shaft 6 of which is mounted centrifugal regulator 7. The reduction gear, generator and supporting device have a common body, into which the two-stage worm orientation mechanism 8 is also built. Two wind roses are located in front of the windmill. The support bearing device 9 is the same /117

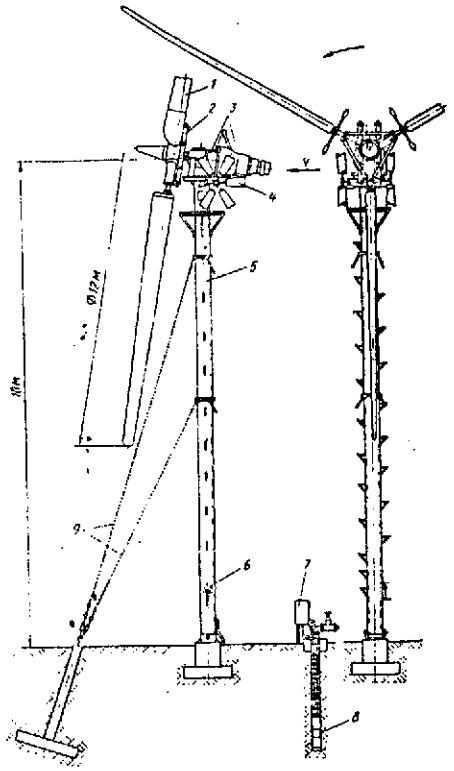

Fig. 80. Overall view of the "Sokol" machine.

as in unified machines. The tubular tower (metal or reinforced concrete) has a hinge at the bottom.

The motor works in the following manner. Before the beginning of operation, the blades are automatically set at the optimal starting angles (about 40°) and as the angular velocity of the windmill increases, they rotate to the working angles (13°). The springs of the moment regulator, enclosed in the housing (see Fig. 42), have a preset tension which corresponds to the nominal moment M_n. If the moment of resistance of the load M_r becomes greater than M_n, the hub of the windmill, overcoming the force of the spring, turns relative to the shaft, and the gears of the roots (see Fig. 81), rolling along the shaft gear, turn the blades so that the moment developed decreases. In the reverse situation, the hub is rotated by the action of the springs.

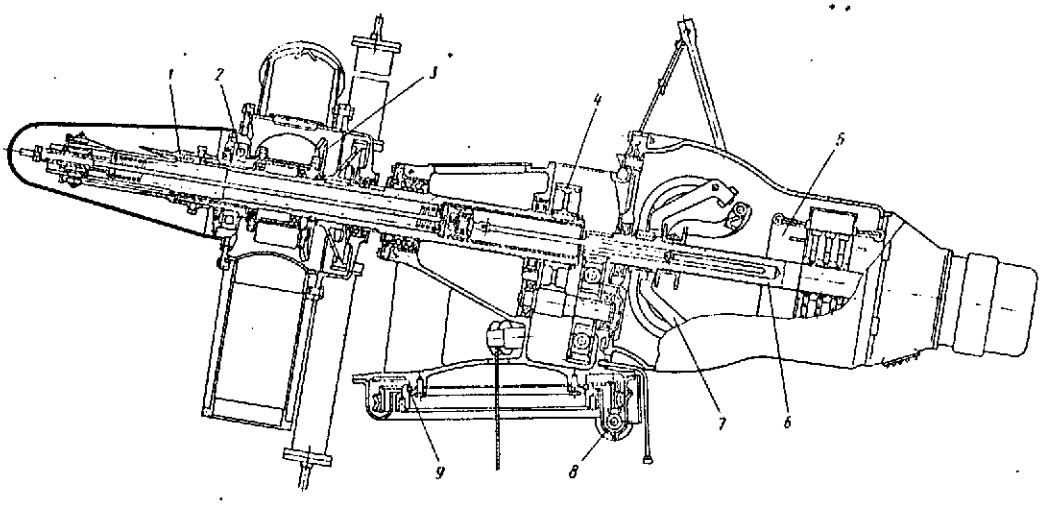

Fig. 81. Head of the "Sokol" machine: 1. main shaft; 2. bearing; 3. gear; 4. reduction gear; 5. generator; 6. generator shaft; 7. centrifugal regulator; 8. orientation mechanism; 9. support bearing.

At $M_r < M_n$, the rotation frequency of the windmill begins to increase and centrifugal regulator 8 (see Fig. 42) comes into operation. Sleeve 1 of the regulator is connected with pull rod 6, passing inside the main shaft. Overcoming the resistance of spring 2, this sleeve is shifted to the right by the force of the regulator, and, through rollers and a sleeve with inclined slots connected with gears, rotates the blades so that the original rotation frequency is reset.

The motor is stopped with winch 11, by pulling on a cable connected with a lever. Having compressed spring 2, the blades turn so that M_{mo} becomes equal to zero. Depending on the velocity of the wind, the machine has the following characteristics:

Wind velocity in m/sec	4	5	6	7	8 and higher
Power in kW	0.8	1.6	4.5	7.3	12
Productivity in m^3/h:					
at H_Σ = 50 m	--	5.9	11.3	14.1	16
at H_Σ = 100 m	--	4.6	9.7	12	15

§ 3. Foreign Wind Machines

By the end of the 1960's, more than 600,000 machines, primarily low-speed ones, were being used. In Australia alone, they number over 250,000. The most interesting are the machines produced in France, USA, England, Australia and FRG, which are designed for pumping water, charging storage batteries and illumination of objects, cathode protection of pipelines and for some special purposes. Research and tests of sample models of large wind-electric stations and machines (P > 15 kW) for work in power-producing systems or together with non-wind installations is being conducted. Requirements outlined for the machines include operation at low wind velocities (3-3.5 m/sec); high reliability, including operation during storms, which necessitates the use of corresponding automatic systems of regulation and /118
limitation of rotation frequency and power; simplicity of tuning and operation; high technical lever of construction; unification of components, and, in connection with this, simplicity in organization of production; and high economy, allowing wind machines to compete with thermal and electrified installations in cost of energy produced.

The most widely used low-speed pumping installations are the "Hercules" (England) and the "Aeromotor" (USA). The motors are mounted with piston single- and double-action pumps. From a From a windmill with a diameter of 1.8-6 m, having 8-12 blades, the rotational moment is transmitted through a crankshaft-connecting rod mechanism and a vertical rod to the piston. This layout reflects one of the tendencies of foreign wind technicians, which consists of primary utilization for work with low-speed

motor pumps, since, in the opinion of foreign specialists, in zones having a low wind intensity ($\bar{v}_a \leq 5$ m/sec), the effectiveness of high-speed installations is lower than low-speed ones[1]. The rotation frequency of the windmill and the power of these machines is limited by the output of the windmill due to the wind through an eccentric or lateral plane, and orientation of the head is achieved using a tail. The machines are simple in construction and relatively cheap, reliable in operation and have a high coefficient of efficiency.

The towers, with a height of 4.5-18 m, are assembled out of standard elements (angle irons), which provides the possibility for the actualconditions to select the required support. Depending on the diameter of the windmill, the rod accomplishes 17-45 strokes per minute respectively with H_Σ = 120-30 m and d_n = 140-375 mm. The productivity of a machine having a power of 0.3 kW with H_Σ = 30 m is equal to 0.5 m^3/h; a motor with a power of 3 kW raises 1.5 m^3/h of water from a depth of 120 m and up to 5 m^3/h with heads of 45-60 m.

Part of the motors are mounted on water head towers with a capacity of 4-19 m^3 (Fig 82), which decreases capital expenses on the installations' equipment.

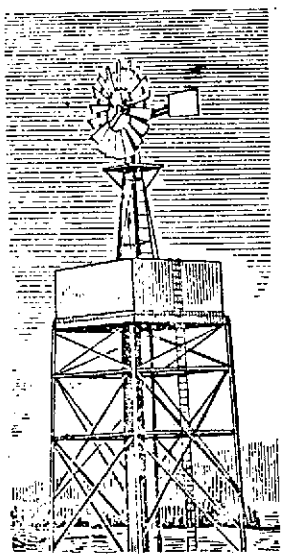

Fig. 82. Wind motor on water head tower.

A feature of the "Hercules" installation is that in the absence of wind, it is possible to rapidly transfer to manual pump drive. For this, the rods of the motor and pump are disconnected, removing an insert.

Pumping wind machines of smaller dimensions (N ≤ 1.5 kW) put out by the Dempster (USA) and Mettner (Australia) firms have a similar layout. Dutch machines with a power of up to 12 kW and medium-speed have four wide rigidly fixed blades, work with a piston pump, and are used for providing water, and raising water during drainage and reclamation. In France, along with "Ideal" low-speed machines, having a power of 0.3-3 kW, with mechanical drive and calculated for lifting0.6-2.5 m^3/h of water from a depth of up to 40 m, the "Noyrpic" three-bladed high-speed electrical pumping machines with a windmill diameter of 8 and 13 m with centrifugal and propeller pumps

[1] Developments performed in the USSR show that this opinion is not justified in all cases (see Chapters 4, 6, 11, 12).

are being serially produced. With a wind velocity of 5-6 m/sec, the machine with a windmill diameter D_{wm} = 13 m raises from 1 m^3/h of water at a head of 50-60 m up to 20 m^3/h with a head of 2-3 m.

In order to improve the work and increase the power of a motor, a housing-fairing with ribs, which increases the velocity of the stream rushing against the turning blades, is installed in the front part of the windmill of some French machines. The windmill is fitted onto the shaft of a synchronized generator, forming the head. One of the electric machines automatically stops as soon as the pump begins to feed saline water.

In the English motor of the Betti firm, a tubular windmill frame is used. By comparison with a frame made of angle steel, the use of a tubular frame decreases the weight of the windmill, makes it stronger, improves the streamlining of the load-bearing grid and decreases losses in power. The gears of the drive mechanism have a number of holes located at various distances from the axis of rotation. Moving the connecting rod pins in the corresponding holes, the piston stroke is changed depending on H_Σ and the required productivity.

These motors are assembled with towers 7-21 m high. To increase longevity, many components (the tower, tail, head, windmill and others) undergo hot galvanizing. The rigidity of blades in low-speed motors is increased due to the utilization of brackets on the leading edge of the blade. For blades of high-speed machines, controlled by a centrifugal or centrifugal-hydraulic regulator, reinforced plastics or duralumin are widely used. In electric machines, the synchronized generators have a voltage of 220-380 V, and with P < 0.5 kW, u_{nom} = 24-32 V.

The Indian WP-2 12-bladed machines (D_{wm} = 5.5 m; P = 2.5 kW) have a head without a reduction gear. The windmill of these machines is fastened on a crankshaft with which the rod is set /120
into motion. To improve working rates, differential plunger heads are used instead of simple action piston pumps. High values for the coefficient ξ (up to 0.37) guarantee good aerodynamic parameters of the low-speed windmills.

A great deal of attention is devoted in foreign practice to selection of the size of the machine, inasmuch as its cost increases essentially with an increase in power. Thus, the distance in the price of "Climax" installations (England) with a power of 1 and 1.5 kW approaches 40%. A tower 6 m high is 30% cheaper than a tower 12 m high.

The D-10 "Algayer" electrical machine (FRG) with a power of 6 kW is assembled with a submerged pump and has three blades controlled through a hydraulic system by a centrifugal regulator. At high wind velocities, the motor automatically stops.

The head of this machine (Fig. 83) has a two-stage reduction gear in a single body with a generator of direct or alternating current (n_n = 1500 rpm) and one wind rose. The support is a tube 10 and 18 m high with two stages of guy wires.

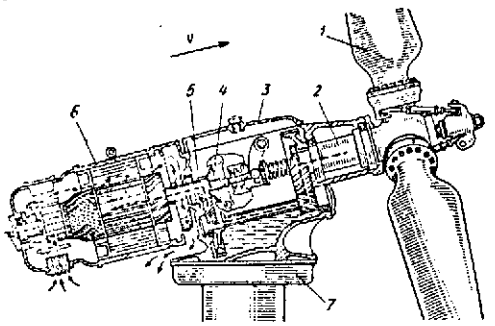

Fig. 83. Head of the "Algayer" wind machine: 1. windmill; 2. main shaft; 3. housing; 4. regulator; 5. reduction gear; 6. generator; 7. head support.

For improvement of machines' characteristics, pumps with a low starting moment and a minimum number of sliding parts are being developed. A bladeless pump was created in Japan for combination with a high-speed motor.

Electrical charging machines are simplified to the maximum: they are without reduction gears and do not have systems for regulating rotation frequency and power. The blades in windmills of many machines are rigidly fastened. Direct current generators without relays and voltage regulators /121 are used on these machines. The machines are used for charging storage batteries, cathode protection, providing power for beacons, and in a number of regions of the USA, also in citrus plantations: turning, they mix the cold layers of air near the earth with warm upper layers, preventing the fruit from freezing.

American firms are putting out two- and three-bladed machines with generators with a power of up to 1 kW, having permanent magnets, and in the FRG, they are putting out the WL-250 and WL-500 with a power of 0.2-0.5 kW (D_{wm} = 2.5 and 5 m). A windmill having Z_n = 7-8 is fastened onto the shaft of the generator. Blades of pressed fiberglass are rotated by a centrifugal regulator in the direction toward a reduction in angles of incidence. With an increase in angular velocity, the blades, due to the effect of centrifugal forces, move somewhat radially, overcoming the forces of the springs. A carrier moves along a spiral slot in the hub, rotating the blade.

A number of motors are started into operation under optimal conditions, since the blades are automatically set at high angles. For easing starting at low wind velocities, an automatic centrifugal sleeve was used, but it has been set aside due to complexity of the machine and a reduction in its reliability.

For convenience in transportation, assembly and servicing, the head is installed on a tubular telescoping mast so that it

can be raised or lowered in a few minutes. An alternating current voltage of 110 V is fed into a transformer, and then is rectified for charging storage batteries.

The French "Siamo" charging machines with a power of 1-5 kW are calculated for operation at a temperature of down to -70°C and a wind velocity of up to 70 m/sec. The windmill with the front fairing has two or three blades made of fiberglass or aluminum and fastened onto the shaft of the direct current generator (u_n = 24/36 V and 110/180 V), which is installed on a metal or reinforced concrete tower. The circular empennage of the tail is carried at a great distance from the plane of rotation of the windmill.

During recent years, a further increase in the speed of machines has been noticed. In the FRG, a superfast motor (Z_n = = 12-16) of the "Bauer" system has been created, with one blade balanced by a counterweight -- the generator, connected with the axis of roation of the windmill through a reduction gear. In order to eliminate vibration and improve self-orientation of the head, the plane of rotation of the windmill is located at a significant distance relative to the axis of the head. The power of the machine is limited by changing the angle of blade setting. To stop the machine, the axis of rotation of the windmill is moved into the vertical position.

Increation of high-power wind machines, primarily for operation with power-producing systems, there is a striving to reduce weight and expand the range of working velocities. A synchronized generator is installed on the head and driven into rotation by the /122
windmill through the reduction gear. The station works in parallel with an electric network or thermal installation.

In the USSR, grouped wind-electrical stations made up of 10-15 machines of 25-30 kW each, working for a common consumer are considered to be more economic. Abroad, preference is given to single-machine stations having a large diameter. The most powerful station (P = 1250 kW; D_{wm} = 53 m) was constructed in the USA; at the present time, the biggest (P = 650 kW) is a French station (Fig. 84). In the Philippines, a design for a wind-electrical station with two windmills moving on a common tower (D_{wm} = 50 m; P = 3000 kW) has been finalized.

The English "Crenfield" wind-electrical station (D_{wm} = /123
= 10 m; P_w = 10 kW; v_w = 8.3 m/sec; n_{wm} = 103 rpm) has a tower 12 m high made of sheet steel. The station is made up of several reinforced blocks and is moved with two truck tractors. The three-phase low-revolution generator has a voltage of 415 V (Fig. 85).

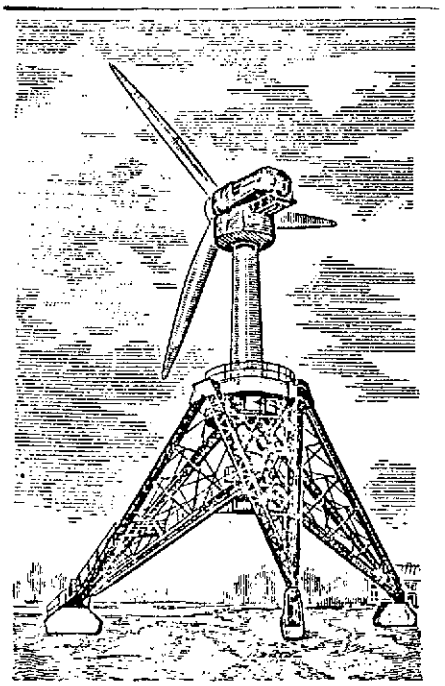

Fig. 84. French wind-electric station with a power of 650 kW.

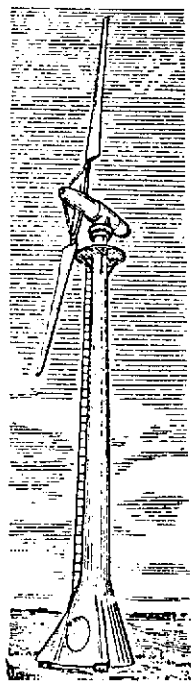

Fig. 85. English wind-electric station with a power of 10 kW.

A station with a power of 100 kW (with $v_w \approx 10$ m/sec), constructed on the Orkney Islands (Great Britain), has a three-bladed windmill (D_{wm} = 24 m) with a centrifugal-aerodynamic regulator. The orientation system of the head is controlled by a vane device with electrical drive.

Foreign wind-electric stations can be divided into two groups according to layout: 1) with direct drive of a three-phase generator from the windmil and 2) with a standard generator and step-up transmission. Regulation of the windmill's rotation frequency is accomplished by merans of turning the blades, controlled by a direct action centrifugal-aerodynamic regulator or through hydraulic drive; orientation of the head is achieved using a wind rose or electrical drive, controlled by a vane.

The original layout for a wind-electrical station with pneumatic drive was proposed by the French engineer Andro and used by the English in the wind-electrical station with a power of 100 kW (D_{wm} = 24 m). The pneumatic drive possesses high flexibility and the wind-electric stations does not fear overloads. The principle of its operation is based on the fact that during rotation of hollow blades in channels, a suction is created and air passes through the tower upward from below with a high velocity. The generator and air turbine are installed in the lower part (Fig. 86).

There exists a distance at which it is economically unfeasible to provide power, required in small quantities, to objects. Even in the presence of electrical power, it is more economic in some sections (for instance, at field wells) to use wind stations than to bring in an electrical line. Wind installations are especially effective in deserts, semideserts and reclaimed zones.

With $\bar{v}_a$ = 4.5 m/sec, the relationship of expenses for raising water with a pump

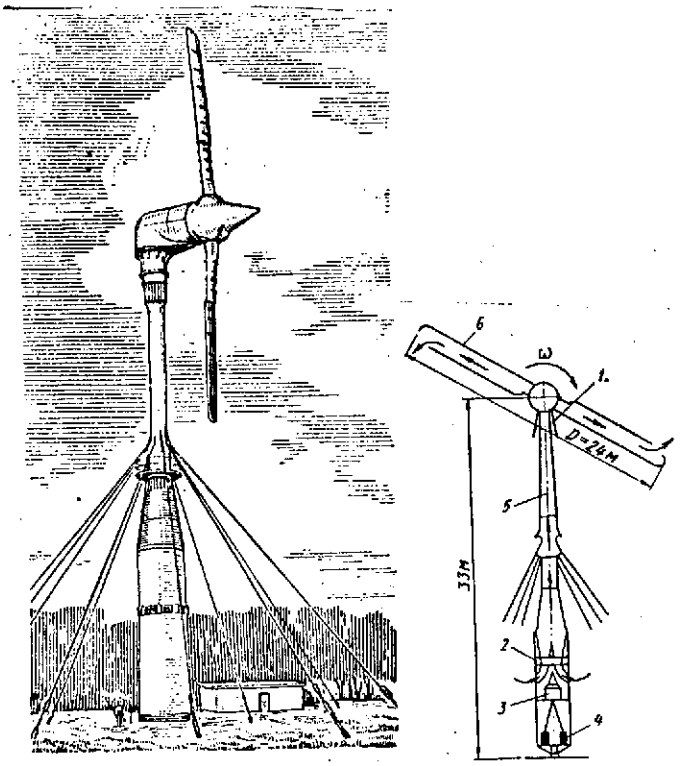

Fig. 86. Wind-electrical station of the Andro system: a. overall view; b. diagram; 1. rotating head; 2. turbine; 3. generator; 4. control apparatus; 5. hollow tower; 6 windmill with passages in blades.

with electrical, gasoline and wind motors is equal to 1 : 10.4 : 0.16, i.e. /124
using wind motors, expenses for raising water are reduced by six times, although its cost is higher. Experience in the FRG in utilization of a set of eight D-10 machines for reclaiming marshy areas showed that the cost of 1 kW h of electric power does not exceed 3.5 kopecks with an average provision of 86.5% of the total electrical consumption; the remaining 13.5% is brought in from a reserve network. According to data of economists in the FRG, the cost of 1 kW h of power developed by charging machines, even with low productivity is 12-20 kopecks in regions with $\bar{v}_a$ = 4-4.5 m/sec. When providing power from a wind electric installation to consumer located at a distance of 0.8 km, the cost of 1 kW h in England is 3-9 kopecks,

depending on $\bar{v}a$.

The technical characteristics of foreign wind machines is given in Table 16.

TABLE 16. TECHNICAL CHARACTERISTICS OF SOME FOREIGN WIND MACHINES

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(a) Наименование агрегата, страна	D_{wm} in m	(b) Число лопастей	n_{wm} in rpm	v_0 in m/sec	v_{wm} in m/sec	N_{wm} in kW	(c) Автоматические системы: регулирования (d)	ориентации (e)	h_{op} in m	G_{na} in kg	Type of pump	H_s in m
Pumping machines												
Climax (England)	2,4—4,2	18	100—60	3—3,5	8	0,4—1,4	Output of windmill due to wind	By the tail	Up to 15	—	Piston	100
Dempster (Australia)	1,8—4,2	24—15	90—70	3,5—4	8	0,3—1,4	Same	»	Up to 15	300—700	»	80
Aeromotor (USA)	1,8—6,0	18	90—50	3.3—3.7	6,3—9,0	0,3—2,5	»	»	6,6—18	261—1450	»	36—175
Hercules (England)	1,8—5,5	18	90—60	3,7—4,0	8—9	0.3—2	»	»	5.7—18,5	—	»	До 100
Ideal (France)	2—6,1	18	—	3,5	8	0,3—2.5	»	»	—	—	»	До 100
Likkergard (Denmark)	7—18	18	—	4	8	6—16	»	»	Up to 15	—	»	150
US Wind Engine (USA)	2.4—3.6	18	100—70	3,5	8—9	0.4—0,9	»	»	8—16	—	»	75
Noyrpik France)	8—13	3	—	4	9	4—10	(f) Поворотом лопастей при помощи центробежного регулятора	»	8—10	—	(g) Электрический центробежный или пропеллерный	40
Algayer (FRG)	10	3	150	3,5—4	9	6	Same	with wind roses	10 & 18	1340—1700	(h) Электрический центробежный	70
Low-power charging electrical machines												
Jacovs (USA)	4.3	3	226	3,5	8	1,5	(i) Поворотом лопастей	By the tail	7—10	—	—	—
Freelight (England)	1 8—4	2	500—1000	4	8	0.15—1,0	(j) Без регулирования	»	5—8	—	—	—
Noyrpik (France)	3.5—8	3	200—400	4	9	1—5	Same	»	6—11	—	—	—

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[continued]

[Key on following page]

Table 16 continued

(a) Наименование агрегата, страна	D_{wm} in m	(b) Число лопастей	n_{wm} in rpm	v_0 in m/sec	v_{wm} in m/sec	N_{wm} in kW	(c) Автоматические системы: (d) регулирования	(e) ориентации	h_{op} in m	G_{na} in kg	Type of pump	H_z in m
Wind-electric stations												
Andro (England)	24.4	2	96	5	13.5	100	(k) Поворотом лопасти при помощи гидросистемы	(l) Электрическим двигателем при расположении ветроколеса за башней	30.5	40 000	—	—
S.E.A.S. (Denmark)	24	3	30	5	15	200	(m) Саморегулирующийся профиль лопасти	(n) Электродвигателем, управляемым флюгером	25	34 000	—	—
Society of Electricians (France)	21.2	3	56	5	12	130	(i) Поворотом лопастей	with wind roses	20	—	—	—
Society of Electricians (France	30	3	47	6	12	600	(k) Поворотом лопастей при помощи гидросистемы	Same	33	—	—	—
R. Smith Harley (England)	15.2	3	75	5	18	100	(j) Без регулирования	»	10.5	—	—	—

Key:
(a) Designation of machine, country
(b) Number of blades
(c) Automatic systems
(d) Of regulation
(e) Of orientation
(f) Rotation of blades using a centrifugal regulator
(g) Electrical centrifugal or propeller
(h) Electrical centrifugal
(i) By rotation of blades
(j) Without regulation
(k) By rotation of blades using a hydraulic system
(l) With electric motor with location of windmill behind tower
(m) Self-regulating blade profile
(n) With electric motor, controlled by vane

PART 3. COMBINATION /128

Chapter 7. Combination of Wind Motors with Working Machines

The combination of a wind motor with a working machine -- a pump, generator, mill -- has a number of peculiarities by comparison with combination of motors which are usually used in heat, hydraulic and other installations. This is first of all connected with the specifics of a wind motor and the difference between its characteristics and operating rates from similar indices of, for instance, a heat motor. We will examine these peculiarities.

Energetic, kinematic and other calculations during combination are as a rule conducted for the nominal rates corresponding to optimal characteristics of a motor (maximum coefficient of efficiency, minimum fuel consumption and so forth). In a wind motor, there is no nominal rate such as this. Instead of this, a rate is isolated which corresponds to a determined wind velocity ($v_{n.reg}$), at which the regulator begins to limit the power developed by the wind motor. This rate, which is called the calculated rate, corresponds to the established power of the motor. Most frequently, the value of $v_{n.reg}$ is selected with consideration for conditions of obtaining the maximum development of energy by the machine in a certain zone or the minimum adjusted costs.

The second feature consists of the fact that the rotation moment of the windmill constantly changes as a result of changes in wind velocity and rotation frequency of the windmill. The range of possible changes in the rotation frequency is great, a wind motor can be most effectively combined with a machine whose characteristics, primarily $\eta_{tot}(\omega)$, change smoothly or remain constant. The absolute size of the coefficient of efficiency has a lower value in this case. With this, it is important that the work of the wind motor at changing rates is not reflected in the quality of continuity of the technological process.

Thus, in several types of silage cutters, the working organs mill the mass poorly at $\omega < 0.7\omega_w$. Vibration pumps sharply decline in productivity and do not always operate stably in the zone of low oscillation frequencies, which are proportional to the rotation frequency of the windmill.

The third singularity is connected with the characteristic of change in the moment characteristic of the moment characteristics of the windmill: the higher its speed, the greater the difference in the value of the relative moments of the windmill at the optimal and beginning rates. In a high-speed windmill, the moment M_o during starting is usually several times smaller than the calculated one (see Fig. 24). Therefore, not only the moment /129 of load during the initial period, but also its gradient $\partial M_s/\partial t$ must be, inasmuch as possible minimal. In combination, the working

rate characteristics are selected so that curves M_{mo} (v, t, ω) and M_s (v, t, ω) pass as closely as possible to each other.

One of the tasks of combination is selection of the type and layout of the windmill, head, drive and working machine, and provision of the interrelationship of their characteristics with each other.

The task of optimal combination of a wind motor with a working machine consists of combining these different devices, using a drive, into a single unit for effective combined operation, providing the least expenses, the required longevity of the machine, reliable execution of the technological process and convenience in operation.

Until recent times, recommendations on combination were based on adaptation of working machines to wind motors already existent [35]. The primary goal of combination consisted of achieving the maximum load for the windmill at a single (calculated), but usually arbitrarily selected rate. With this, the cost indices were not considered. Consequently, the task of optimal combination in a complex could not be solved.

Considering the singularities of a wind motor which create difficulties greater than usual during combination, we will formulate the major requirements for elements of the machine.

The windmill must provide initial operation of the machine at a minimum wind velocity v_0, equal to the optimal value (see Chapter 4), for which the aerodynamic moment developed by the windmill during all rates of acceleration must be greater than the total moments of resistance created by the drive and loading mechanisms. The system for regulating rotation frequency and power of the windmill must provide reliable protection of the machine from overloads and overwearing, as well as operation with a certain evenness in rotation frequency.

Thus, when working with a piston pump and belt lift, velocity ω_{max} must not exceed ω_n by more than 40%. In pneumatic machines, unevenness of rotation does not have any essential meaning and it is important only to protect the windmill from overwearing with high wind velocities. The value of $\omega_{wm\ max}$ is usually limited only be conditions of strength of the windmill and compressor. This is also true in cases where there is a mechanism with an infinitely adjustable gear ratio in the drive system between the windmill and the working machine.

The operation of wind-electric machines with synchronized generators at a current frequency which differs greatly from the nominal one deteriorates the characteristics of the electric motor, pump or other working machine and decreases the coefficient of

efficiency. In the combination of wind motors with direct current generators, the range of allowable changes in rotation frequency can be expanded up to ±30% of ω_n. /131

The high-speed windmill with rotating blades controlled by a centrifugal-aerodynamic regulator, which simultaneously provides optimization of the starting and acceleration rates, answers many of the requirements listed. The drive must provide transmission of the power (moment) with a high coefficient of efficiency within a wide range of change in rotation frequency of the driving link, temperature of the surrounding medium and in the presence of a large quantity of dust and abrasive particles in the air; the minimum numbers of transformations of one form of motion or energy into another; and the possibility for installing the motor in the majority of cases at some distance from the source of water or power consumer.

The mixed drive satisfies these requirements in full measure, but depending on the type of working machine, it is also possible to effectively use drives of other types. In particular, when combining with a generator, a mechanical drive has the best characteristics, especially if the generator and reduction gear compose a single unit.

The working machine must reliably and stably operate at a constantly changing angular velocity, have a low initial moment of resistance, and a sloping external characteristic η (ω, N_{load}) and $M_s(\omega)$. In units of high and medium power, the possibility for rapidly changing over to driving the machine from a duplicating motor must be provided.

For pumping machines, depending on the type and characteristics of the water source, additional requirements arise for the water pump. Thus, for low-yield and seepage wells, it must have laminar water collection, so that muddying of the water does not occur. Of the known water-raising devices, the screw-type, pneumatic, sucker-rod, vibration and centrifugal pumps satisfy the majority of the requirements (Fig. 87).

For wind-powered units, machines should be used in which the quality of processing products is high in a wide range of angular velocities of the working organs and at constantly changing velocities. Grinding mills, hammer crushers and circular saws should be included in these.

In wind-electric machines, it is advisable to use generators with permanent magnets and synchronized generators with excitation from a permanent source of energy, and, in a number of cases, also nonsynchronized generators [4, 46].

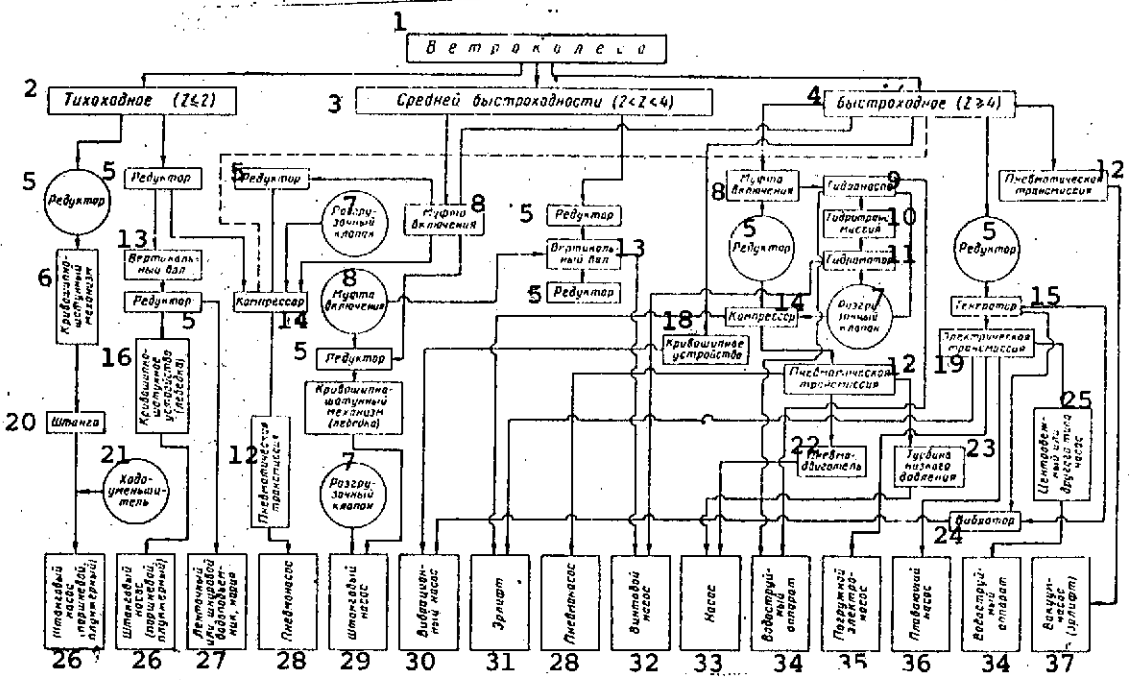

Fig. 87. Variations of principal layouts of water-raising wind machines.

Key:
1. Windmill
2. Low-speed (Z < 2)
3. Medium-speed (2 < Z < 4)
4. High-speed (Z ≥ 4)
5. Reduction gear
6. Crankshaft-connecting rod mechanism
7. Relieving valve
8. Clutch
9. Hydraulic pump
10. Hydraulic transmission
11. Hydraulic motor
12. Pneumatic transmission
13. Vertical shaft
14. Compressor
15. Generator
16. Crankshaft-connecting rod device (winch)
17. Crankshaft-connecting rod mechanism (winch)
18. Crankshaft device
19. Electric transmission
20. Rod
21. Stroke-reducer
22. Pneumatic motor
23. Low pressure turbine
24. Vibrator
25. Centrifugal or other type of pump
26. Rod pump (piston, plunger)
27. Belt or band water lift, water wheel
28. Pneumatic pump
29. Sucker-rod pump
30. Vibration pump
31. Air lift
32. Screw pump
33. Pump
34. Water jet apparatus
35. Submerged electric pump
36. Floating pump
37. Vacuum pump (air lift)

§ 1. Utilization of Mechanical Drive

Mechanical drive is usually used when combining wind motors with low-speed machines (mills, silage cutters, feed crushers, various types of pumps). The methods for selecting drive characteristics for these machines and methods for combination are sufficiently well developed and laid out in the corresponding literature [35]. In recent years, mechanical drive is also being ever more frequently used in combinations of high-speed motors with screw and inertia pumps, and also with belt water lifts. /132

Combination with screw pumps is possible with different motor speeds, but machines of medium speed are more effective, in particular due to the reduction in velocity v_0 (Table 17). However, it is also possible to achieve the required working rates of the machine (Fig. 88) when combining a high-speed motor ($Z_n \geq 3$) with a pump having a relatively small screw diameter. In calculations of machines, generalized characteristics of the type shown in Fig. 89 are used.

TABLE 17. SOME CHARACTERISTICS OF MACHINE WITH SCREW PUMPS

Windmill parameters												
D_{wm} in m	$\bar{M}_0$	$\bar{M}_{max}$	z_n	i/η	$n_{ск}$ (a) в об/мин при $v=7$ м/сек	M_0 в н·м при v в м/сек (b)		
						3	4	5
6	0.015	0.07	6.6	2	270	2	3.5	6
4	0.022	0.12	4.2	3	160	3	5.2	9
3	0.3	0.44	1.2	12	80	19	3.7	52.5
4	0.022	0.105	4.5	3	180	3	5.2	9

Характеристики нагрузки (c)											
n_n в об/мин при d_n в мм (d)			M_{0r} в н·м при d_n в мм (e)			Передаточное число привода i_n при d_n в мм (f)			v_0 в м/сек агрегата при d_n в мм (g)		
16	25	36	16	25	36	16	25	36	16	25	36
800	275	180	10.5	9.8	11	3	1	0.7	6.5	6.35	6.55
800	275	180	17.5	16.5	17	5	1.7	1.15	6.5	6.4	6.5
800	275	180	35	34	35	10	3.5	2.25	4.15	4.0	4.1
800	275	180	15.5	14	15	4.5	1.5	1	6.1	6.0	6.1

Key: (a) n_{wm} in rpm with v = 7 m/sec
(b) M_0 in n·m with v in m/sec
(c) Characteristics of loading
(d) n_n in rpm with d_n in mm
(e) M_{0r} in n·m with d_n in mm
(f) Transmission ratio of drive i_n with d_n in mm
(g) v_0 in m/sec of the machine with d_n in mm

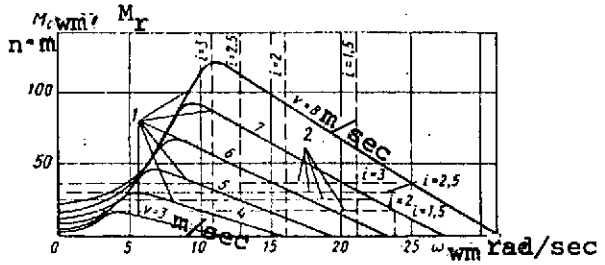

Fig. 88. Combination of characteristics of a windmill (D_{wm} = 4 m; i_ℓ = 3) and a screw pump (d_n = 25 mm) with various drive transmission ratios: 1. $M_{wm}(\omega_{wm})$; 2. $M_r(\omega_{wm})$

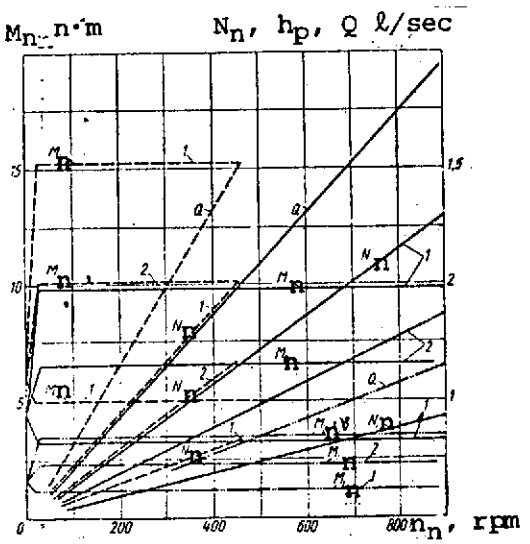

Fig. 89. Characteristics of single-screw pumps of the design by D. D. Savvin: 1. H = 30 m; 2. H = 20 m; 3. H = 10 m; broken lines - d_n = 36 mm; solid lines - d_n = 25 mm; broken-dotted lines - d_n = 16 mm.

The machine is simple in construction and in operation, and therefore high qualifications of the servicing personnel are not required. This is also enhanced by the fact that the pump has small dimensions and weight, operates stably in contaminated water, and the elastic rubber ring resists the action of sand well. Due to slight leakage of liquid between the channels of the screw and the ring, the characteristic $\eta(H_\Sigma, \omega)$ is sliding; the feed of the pump is practically proportional to the number of revolutions and depends littled on the total pressure. The /134 coefficient of efficienty of the majority of pumps is 0.5-0.55, and pumps with a new geometry of the screw pair have a higher coefficient of efficient (η = 75-80% at H_Σ = 30 m). A singularity of these pumps is the property of reversibility; when the fluid is fed into the pump under pressure, it can work as a hydraulic motor, which provides the possibility for using converted pump stages in a machine using hydraulic drive.

To improve the starting rates of the machine, it is necessary that there is water in the pump, since the moment of friction increases with a dry ring. After stopping the motor, however, a large column of water should not be left above the pump, since this increases the moment M_{or}. Water is removed from the pressure mainline using the drain pipe or relieving valve.

One of the tasks during combination is overcoming difficulties caused by the fact that the initial moment of resistance from the force of friction of the screw on the ring, the column of water and in the bearings of the vertical shaft is relatively high. This moment depends little on the total pressure, and the gradient of increase of the moment during transfer of the machine to the calculated working

rate is significant. Therefore, the rates of start-up and acceleration are the most difficult. At the same time, it is necessary to fulfill conditions emanating from the results of calculations for optimization, and in particular to obtain $v_0 = v_{o\ opt}$.

The required average hourly productivity of a machine is determined by the water consumer. For a certain motor, the use of a smaller diameter pump does not improve the indices of the machine, since with a decrease in the diameter of the pump, an increase in rotation frequency is required. The transmission ratio of the drive i_n increases and the moment of resistance presented changes little, but with an increase in i_n, the coefficient of efficiency is somewhat decreased.

To improve the operational indices of the machine, it is expedient to relieve the motor during starting, using a special valve in the pump or a mechanism for automatically engaging and disengaging the transmission and pump, which is controlled by the centrifugal regulator of the windmill. The principle of operation of the mechanism has much in common with the principles for relieving a compressor and therefore the operating rates of the motor with a screw pump can be calculated according to the methodology used for computing pneumatic machines (see § 2 of this chapter). The operation of the valve is based on its capability to release part of the water into the source when the pump is feeding little, i.e. during the period of the windmill's acceleration, and also to reduce the initial pressure in the water column. At calculated rates, the valve closes and all the water flows into the pressure mainline.

In calculating a mechanical drive, it is necessary to consider the possibility for the arrival of overloads in the transmission when the device connecting the rotating windmill with the stationary shaft of the pump is engaged. Considering that the different ΔE of the reserves in kinetic energy of the system before and after engagement is transformed into potential energy P_d of deformation of a shaft of length l_s and transforming to the Karno theorem for rotational moment, we obtain /135

$$\Delta E = \frac{J_{pr} i_n^2 J_{dn}}{2(J_{pr} + i_n^2 J_{dn})} \omega_{pr}^2 = \frac{M_{max} \psi_{s\ max}}{2} J, \qquad (104)$$

and the maximum moment in the transmission under shock

$$M_{max} = M_r + M_{ovl} = M_r + \omega_{wm}^2 \frac{G'_{tr} J_{tr \cdot k}}{l_s M_{tr}} \left(\frac{J_{pr} \Sigma i_n^2 J'_{dn}}{J_{pr} + \Sigma i_n^2 J'_{dn}} \right) n \cdot m \qquad (105)$$

where J_{pr}, J'_{dn} are the adjusted moment of inertia in the power and driven parts; $G'_{tr}J_{tr.k}$ are is the rigidity of transmission (or vertical shaft) during torsion; $\psi_{s\ max}$ is the maximum angle of twisting of the shaft.

Usually, $J'_{dn} << J_{pr}$, and therefore the overloads arising during engagement of the machine are not very great. Calculations for a machine with a power of 1.5 kW show that with $H_\Sigma = 30$ m, for a pump with a diameter $d_p = 25$ mm, overloads, by comparison with the nominal moment N_n do not exceed 50%, if the machine is engaged at $n_{wm} = 100$ rpm.

One of the tasks of combination is to decrease the gradient M_r in the starting rate. It is solved using the same velocity pressure valve.

If the required productivity Q_p of the pump is given, its nominal rotation frequency n_p is computed according to the formula

$$n_p = \frac{Q_p}{2.4 \cdot 10^2 T_o e d_p \eta_{vol}} \text{ rpm.} \qquad (106)$$

The power to the pump shaft is

$$N_p = \frac{Q_p \gamma H_\Sigma}{10.2 \cdot 3.6 \cdot \eta_p} \text{ kW,} \qquad (107)$$

where Q_p is the productivity of the pump in m^3/h; T_o is the width of the ring; e is the eccentricity of the screw in m; γ is the specific weight of the liquid; d_p is the section diameter of a screw with a plane perpendicular to its axis, in m; η_p, η_{vol} are the total and volume coefficient of efficiency of the pump.

Determining the optimal value $i_n = \omega_{wm}/\omega_p$, it is necessary to keep in mind that with a change in ω_p up to ω_{pi}, the values Q_{pi} and N_{pi} change:

$$Q_{pi} = \frac{Q_{pi}}{\eta_{rev}} \cdot \frac{\omega_{pi}}{\omega_p} \left[1 - (1 - \omega_{rev}) \frac{\omega_p}{\omega_{pi}} \right] \text{ m}^3/\text{h;} \qquad (108)$$

$$N_{pi} = N_p \frac{\eta_p}{\eta_{rev}} \cdot \frac{\omega_{pi}}{\omega_p} \left(1 - \frac{\eta_{rev} - \eta_p}{\eta_p} \right) \sqrt{\frac{\omega_p}{\omega_{pi}}} \text{ kW.} \qquad (109)$$

In combination of graphs $Q_p(\omega_p)$ and $N_p(\omega_p)$ with moment characteristics of the windmill, losses in the drive are added to M_n. /136

For combination with belt water lifts, motors of medium speed (two- and four-bladed) with values for the initial adjusted moment of resistance of 14-22 n·m are more economic. Usually in such machines the initial moment of resistance M_{oc} of the belt is relatively small, and the coefficient of resistance of the water lift in a certain range of velocities changes little, although its absolute value is low. The low coefficient of efficiency is compensated to a known degree by the simplicity of the devices and their reliability in operation.

It can be considered that the productivity of the water lift Q_{wl} is proportional to the velocity of the belt with $v_b > 2.2$ m/sec, and the minimum (critical) velocity of the belt $v_{b.cr}$, at which water still flows, is equal to 1.4-15. m/sec and increases with a decrease in the wettability of the belt by water. Therefore, the transmission ratio between the windmill and the driven pulley must be selected in a way so that with v_{0opt} the condition $v_b > v_{b.cr}$ is observed. The transmission ratio satisfying this requirement is equal to

$$i = \frac{\omega_{pl}}{\omega_{wm}} \geq \frac{2v_{b.cr}R}{d_{pl}Z_p v_0 k_{sl}}, \qquad (110)$$

where d_{pl} is the diameter of the water lift pulley in m; $k_{sl} = 0.95$-0.97 is the coefficient, considering sliding of the belt along the pulley.

At slight depths (up to 25 m), the power required by the water lift with an increase in velocity v_b increases more intensively than does productivity Q_{wl} (Fig. 90). In the range of belt velocities from 3.5-5.5 m/sec, the specific consumption of energy of the water lift q_{wl} is minimum; at higher velocities the consumption q_{wl} increases, but η_{wl} can also increase somewhat. From the energetics point of view it is important that the rates of minimum specific consumption $q_{wl\,min}$ corresponds to the work of the machine in the range of the most frequently repeated wind velocities.

The characteristics of series-produced water lifts do not fully satisfy requirements for combination with wind motors. This noncorrespondence at low velocities of the wind will lead to a reduction in productivity Q_{wl}, and at high velocities it will lead to an overfilling of the water intake housing and to an increase in the loss of water spilling into the well. Therefore,

in combination with wind motors, water lifts having a low moment M_{or} are used and the water-removing plate is installed on the downward-moving side of the belt [53], which increases productivity Q_{wl} at low belt velocities by almost twice and reduces $v_{b.cr}$. Inasmuch as low belt velocities correspond to the frequently repeated wind velocities of 3.5-4.5 m/sec, the annual output of the machine is increased by 4-6% and the energy-producing characteristics are improved (with $v_b \approx 3.5$-4.5 m/sec; $\eta_{wl} = 16.5\%$). However, at higher and lower belt speeds ($3.0 < v_b < 5.0$ m/sec), the coefficient of efficiency drops sharply (Fig. 91). /137

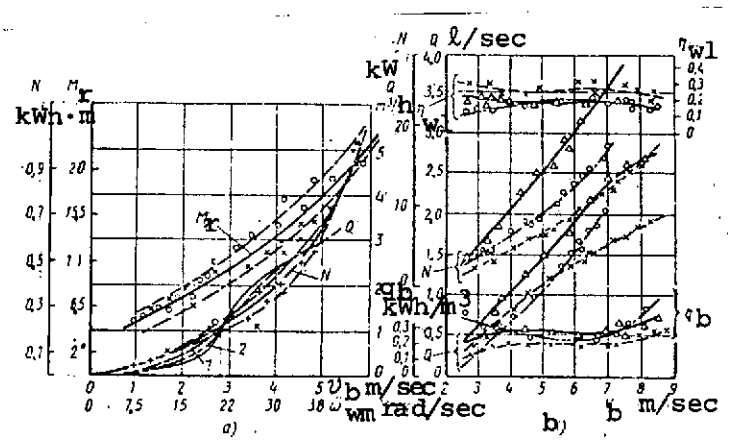

Fig. 90. Characteristics of water lifts: a. with perforated belt having dimensions of 60 x 4 mm with H = 10 m; b. with perforated belt having dimensions of 100 x 4 mm with H = 20 m (solid lines); with a belt having dimensions of 60 x 4 mm (broken lines); with a belt having dimensions of 100 x 4 mm without perforations (dashed and dotted lines); 1. with water-removing plate; 2. without plate.

One of the requirement of optimal combination, caused by the fact that values ω_{wm} and v_b are variables is that a large part of the time the value v_b will change within limits of 3-5 m/sec, when the specific consumption of energy is minimum.

High-speed movie photography showed that at belt velocities close to critical ones, an in the absence of a water-removing plate, a significant portion of the water is not gainfully utilized, since the centrifugal forces are low (Fig. 92). At high belt velocities (7-8 m/sec), all the water is thrown off the belt on the right- and upper part of the pulley.

Without the plate and with a belt velocity of 1.15 m/sec, the feed is almost zero: the water flows back into the well through the hole for the downward-moving side. The plate removes this water and directs it into the drain pipe. With a belt speed of 1.3 m/sec, water is thrown from the belt, as soon as the side entering the body lies against the pulley. On the downward-moving side, a thin, partially separated layer of water is visible. Centrifugal forces at this velocity are smaller than the weight of the drops and the forces of their adhesion to

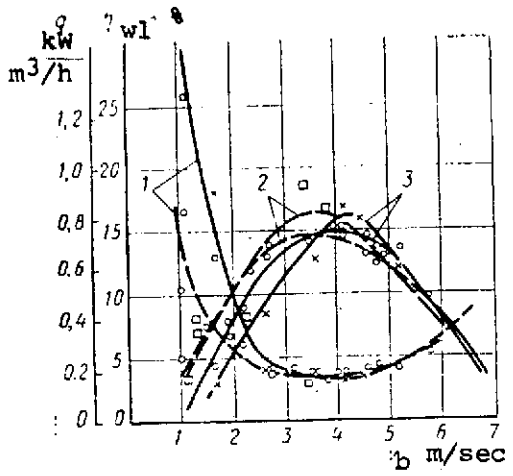

Fig. 91. Energy-production indices of a water lift with belt dimensions of 60 x 4 mm at H = 10 m with a water-removing plate (broken lines) and without a plate (solid lines): 1. $q(v_b)$; 2. $\eta_{wl}(v_b)$ with an increased value for ω_{wm}; 3. $\eta_{wl}(v_b)$ with a reduced value of ω_{wm}.

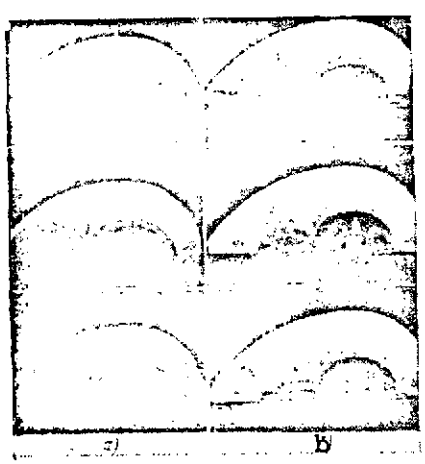

Fig. 92. Trajectory of water particle motion (high-speed cinematography): a. with low values of v_b; b. with high values of v_b.

the belt. Therefore, the plate also plays a large role here, increasing the productivity of the water lift by 2-2.5 times.

An increase in belt velocity /138 changes the character of water feed. Water fills the ever increasing volume of the housing, the effect of the plate is decreased and with $v_b \geq 2.7$ m/sec, the presence of the plate no longer increases the productivity of the water lift.

Designating belt speeds as $V_{b.opt}$, and the corresponding q_{min}, we determine i_{opt} by substituting in expression (110) the value $v_{b.opt}$ instead of $v_{b.cr}$, and wind velocity corresponding to $\bar{E}_{min}$ instead of v_0. Of the two values of the tranmission ratios (i and i_{opt}), the maximum one is selected, considering it the calculated one i_c. The aerodynamic moment of the windmill M_o must be greater than M_{wl} by the value of the moment M_g of the forces of friction in the gears.

The productivity of the machine is dictated by the water consumer or is limited by the yield of the source; with $v_i \geq v_{n.reg}$, is must not exceed the yield. We obtain the parameters of the water lift from formula

$$Q = P_b \frac{1.07 v_b d_{pl}}{R} - 6.33R \text{ ℓ/sec} \quad (111)$$

where P_b is the wetted perimeter of the cross section of the belt in m; R is the coefficient of roughness of the belt (for a rubberized belt, R = 0.37-0.67) [51].

The required diameter of the /139 windmill is

$$D_{wm} = \sqrt{\frac{QH}{0.049 v_{n.reg}^3 \xi \eta_{oa}}}\ м, \qquad (112)$$

where $\eta_{oa} = \eta_d \eta_{mech}$ is the overall coefficients of efficiency of the machine (for the water lifts used, with H = 7-30 m, η_b = 0.11-0.35; $\eta_{mech} \approx$ 0.8-0.85).

An example of combining a windmill (D_{wm} = 4 m; Z_n = 5) with a water lift having a belt 60 x 4 mm (Fig. 93), shows that the machine begins to feed the water at $v_0 \geq 3.3$ m/sec and the rates of its operation will correspond to a coefficient ξ which is close to ξ_{max} within a wide range of belt velocities.

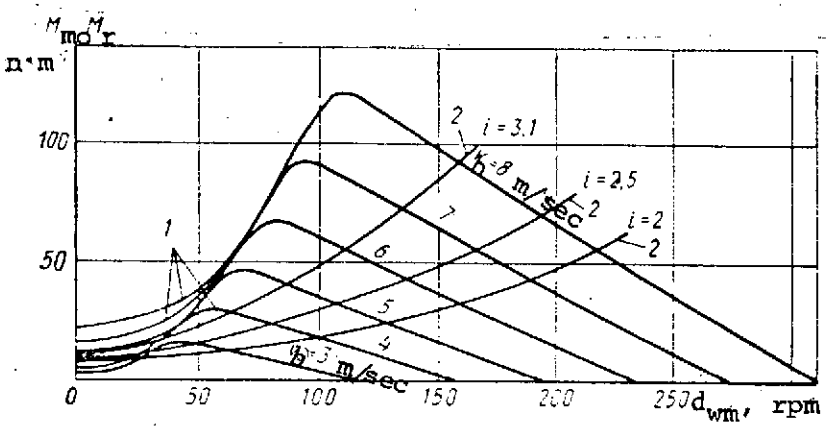

Fig. 93. Combination of a D-4 motor with a belt water lift 60 x 4 mm with H = 10 m: 1. $M_{mo}(n_{wm})$; 2. $M_r(n_{wl})$.

If it is necessary that the machine fulfill other conditions, the transmission ratio i is changed, moving the characteristics of the water lift and the motor relative to each other. Usually, the drive is calculated so that v_b = 3.5-5.5 m/sec, increasing the productivity of the water lift by means of using a wider belt. As is evident from Fig. 93, depending on the tranmission ratio of the drive i, the beginning of operation of the machine corresponds to value v_0 = 3.5; 4.5 and 5.5 m/sec. With $v_{n.reg}$ = 7 m/sec and i = 2.5, the water lift loads the windmill at approximately 70% power. Using removable pulleys, it is possible to set the required transmission ratio depending on the region.

As a rule, when combining using a medium-speed motor, the value i does not exceed 3 due to the relatively low value of M_{or}. At normal operational rates, the moment M_r changes little,

and overloads comprise 20-30%. The range of permissible changes in belt speed Δv_b is relatively great (±30%), in connection with /140 which the requirement for a motor regulator is reduced, simplifying its design. The power required by the water lift changes depending on the number of revolutions so that it is possible to effectively combine the characteristics of a motor and a load within a wide range of wind velocities. Differing from a piston pump, the belt water lift loads the windmill with a smoothly changing moment. In order to decrease slippage of the belt at $v > v_{n.reg}$, the values of ω_{max} of the windmill are limited ($\omega_{max} \leq 1.3\omega_n$) and the forces of cohesion of the belt with the pulley are increased. A decrease in the depth of the pulley beneath water level affects the operation of the machine positively.

For complex utilization of a machine, the possibility for connecting a generator to the transmission is sometimes provided, which allows the characteristics of the windmill and load to be better combined. In this machine, it is necessary to provide rapid disconnection of the generator during periods when the power of the motor is sufficient only for operation with the water lift.

Combination of wind motors with vibration type pumps provides the possibility for effective utilization of both mechanical and electrical drives. The determining factor with this is the pump layout. A mechanical drive is simpler in design and with $H_\Sigma \leq 20$=25 m allows a reduction in expenditures for lifting water.

The singularity of the machines examined consists largely of the fact that the effects of resistance moments of the load are greater than for machines of other types and of the necessity for using higher values of initial wind velocities (v_0 = 4-6 m/sec) and a reduction in operating range (v_0-$v_{n.reg}$).

The characteristics of the pump (high-speed, low value M_{or}) provide the possibility for fulfilling these conditions in combination of the pump with a high-speed windmill. The advantages of this layout are the absence of reduction gears, a high value for η_{mech}, simplicity of design and low metal consumption.

With the pump being directly driven from a crankshaft on which the windmill is fastened, there is a limitation in the height of water lift, since with a tube length greater than 15 m, the reserves in their strength with consideration for dynamic overloads is usually not great. Therefore, for instance, the VB-3T water lift is calculated for wells up to 15 m deep. The reliability of an installation can be increased by using thin-walled tubing of high-strength materials with a specific weight less than that of steel or by using tubing having changeable diameter and wall thickness.

The moment of load resistance M_r overcome by the windmill changes according to a sinusoid period. Moment $M_{r\ max}$ corresponds to the stroke of the pipe upward and to the horizontal position of the crank radius. If the weight of the pipe and pump (G_p) is not balanced, then

$$M_{or} = (G_p + G'_w)r_k + P_f r_c + M_{f.b} \quad n \cdot m \qquad (113)$$

where G_w is the weight of water in the pipe in n; r_c is the radius of crank in m; P_f is the force of friction when withdrawing the pipe in n; $M_{f.b}$ is the moment of the force of friction in the bearings in n·m. /141

In connection with the fact that the pump moves with an acceleration, dynamic forces depending on ω_{wm} arise in the machine. They are summarized with static forces from the weight of the tube with the water and pump. Therefore, it is necessary to compare the moving aerodynamic moment with the moment of summary forces at all operating rates. In order to reduce moment M_r from the weight of the tube with the water and pump, a spring is installed in the drive system, and its force when the radius of the eccentric (crank) is in the horizontal position, balances the weight of the pipe, pump and half the weight of the water in the pipe. The presence of the spring reduces the value for a minimum working velocity of the wind.

The effect of the spring can be taken into account if the expression $(P_{sp} + k_{sp}h_i)r_c \cos \alpha$ is introduced into the right side of formula (113) with a minus sign (where p_{sp} is the force of preliminary tension of the spring; k_{sp} is the rigidity of the spring; h_i is the vertical movement of the water-lifting pipe). The minimum frequency of vibration of the pump must be no less than 10% higher than critical, in which case the forces of inertia are already sufficient to raise water:

$$f_{min} \geq 1.1 f_{cr} \approx \frac{Z v'_0}{27.5 D_{wm}} \quad Hz, \qquad (114)$$

where v_0 is the velocity of the wind at which the windmill begins to turn.

A mechanical drive is also frequently used in low-speed machines with piston pumps, which are simple in design and have a high coefficient of efficiency which depends little on the number of strokes, i.e. on the frequency of rotation of the windmill.

The main task of optimal combination with this layout is the selection of the best correlation between the angular velocity ω_{wm} and the number of double strokes of the piston, and between its diameter d_n and strokes h. This does not usually cause great difficulties, since the initial moment of resistance does not have much effect on the indices of the machine. For set conditions (i.e. for Q, v_{per} and H_Σ), the values of $D_{wm.opt}$ and $v_{n.reg.opt}$ vary within relatively small limits. Velocity v_0 is clearly connected with these parameters and with the values of the transmission ratio. The average moment of resistance of the pump depends little on the number of piston strokes.

Fulfillment of the initial conditions of combination are provided by setting up combined characteristics of the moving aerodynamic moments of the windmill and the moments of resistance of loads so that they intersect at 0_{opt}. Then fulfillment of the conditions is checked so that $M_{r\ max}$ with $v = K_{pis}v_0 = (1.3-1.4)v_0$ is less than M_{mo} of the windmill.

The initial moment of the windmill is equal to $M_0 = \bar{M}_0 \pi R^3 \rho \frac{v_0^2}{2}$. During a gust of wind, moment M_0 will be K_{pis}^2 times greater.

The power at the shaft of the windmill necessary to drive the pump is

$$N_{wm} = \frac{Q\gamma H_\Sigma}{102\eta_p\eta_m} \quad kW$$

where η_p is the overall coefficient of efficiency of the pump; η_m is the mechanical coefficient of efficiency of the windmill.

Since $M_r = N_p/\omega$, then, by changing the sizes of values N_p or ω, it is possible to assure that M_r at v_0 is less than or equal to M_0. During a gust of the wind, $M_{r\ max} = 3.14M_{r.av}$ is used.

The pressure head is always given, and therefore, by varying the value

$$Q = 0.012d_p^2 h n_2 \eta_{rev} \quad \ell/sec,$$

we obtain the needed value for M_{or} at $v_{0\ opt}$. The combination of characteristics is further checked for all other operating rates of the machine. Because of dynamic conditions of the work of the

pump, the number of double piston strokes per minute does not usually exceed 50.

The diameter of the piston and the length of its stroke are standardized and they are selected with consideration for the diameter and yield of the well. By matching the sizes of values d_p, h and n_2, the optimum values for other parameters of the machine are obtained. Experience shows that it is usually more advantageous to decrease h and d_p, increasing the number of strokes. Since the water consumption B_{max} is known, such parameters of the drive are used so that $Q_\Sigma = \Sigma^B$ and $\bar{C}_{ma} = \bar{C}_{ma\ min}$.

The economy of the machine is increased with automatic changing of the piston stroke (proportional to v^2) or the number of strokes (proportional to v^3). With conditions $h_i = h_w(v_i/v_w)^2$ and $n_{2i} = n_2(v_i/v_w)^3$, the windmill will work with a coefficient of $\xi = \xi_{max}$ at all rates. The fulfillment of the first condition requires usage of speed dampers, and the second requires use of automatically controlled infinitely adjustable variators. Due to complexity in design of the machine, installations of either of these types are rarely used.

For combination of a high-speed windmill with a plunger pump, mechanical drive is also used and wide blades are also frequently used. We note that in these machines the coefficient of efficiency of the pump depends on the frequency of rotation of the windmill, i.e. on the velocity of the wind, decreasing with $\omega_{wm} < \omega_{wm.nom}$ (in a piston pump the coefficient of efficiency changes little under these conditions). For high-speed machines, one of the methods for improving the combination layout is use of a relieving valve.

Machines with mechanical drive of a centrifugal pump usually require the installation of two- or three-stage reduction gears and have a relatively high coefficient of efficiency only in a narrow range of velocities. These machines have not been used here, and abroad they have been used only on open water sources, i.e. with low pressure heads. By comparison with wind-electrical machines with submerged pumps, they are preferable in cases where $H_\Sigma < 20$ m and the motor can be mounted above the water source. This decreases the cost of providing water by 10-17%. ON the basis of characteristics Q-N and Q-H with n_p = const, reconstructed for the case n_{wm} = var, the required rotational moments of the windmill, the maximum power of the motor and characteristics of the drive are determined. /143

Considering (with an error of 2-4%) that with a short pipeline, the head little depends on n_p, we write

$$N_i = N_c\left(\frac{n_i}{n_c}\right)^3 \text{ and } M_i = M_c\left(\frac{n_i}{n_c}\right)^2. \qquad (115)$$

With $n < 0.8n_c$, the coefficient of efficiency of the pump falls sharply, and therefore the characteristics of the drive are selected so that with $v = (1.15\text{-}1.2)v_0$, the rotation frequency of the pump is close to the calculated one. Solution of this problem is eased by the fact that the initial moment of resistance of the pump is low and does not usually limit the use of higher speed windmills.

§ 2. Bases of Selecting and Computing a Pneumatic Drive

The use of a pneumatic drive allows reliability and effectiveness of pumping installations to be increased, their metal consumption to be reduced and simplifies control. The possibility of transforming energy using compressed air in several directions allows utilization of one wind motor in a machine with several pumps or other working machines. Installing the motor away from the water source at an elevated place, where the wind rate is more intensive, it is possible to increase the output of the machine by 20-35%.

In conditions where the motor works with its rotation frequency and power constantly changing within wide limits, pumps of some types have low values for their average operational coefficient of efficiency. The rotation frequency of the windmill does not have a great effect on the effectiveness of the work of a pneumatic pump in the presence of a receiver; dynamic losses with this also remain constant. The coefficient of efficiency of a pneumatic pump with H_Σ = const depends only on boundary pressures of the air in the pressure main line and tuning of the distributor device which determines the conditions of compressed air feed.

One of the singularities of combination using a pneumatic drive is caused by the fact that the values for wind velocity at which the windmill begins to turn (v_0), and the machine pumps water ($v_0 = v_{\text{work min}}$) are very close (for machines with centrifugal pumps, v_0 is v_0 = 1-1.5 m/sec).

The primary tasks in combination consist of agreement of /144
the characteristics of the windmill and compressor, in the selection of optimal correlations between the parameters of the compressor, pump and devices controlling the drive, and finally of finding the minimum necessary capacity of the receiver and pipeline section. Depending on the combination layout, the controlling elements of the drive might be a reduction valve, a clutch, automatic device and others.

The use of piston compressors having large initial moments of resistance and requiring good filters and cylinder wall lubrication is not very promising. In order to improve the starting qualities of the machine and its pickup, it is necessary to install

a centrifugal friction sleeve between the windmill and the compressor and to unload the windmill during starting, disengaging the compressor from it. At a set windmill frequency of rotation, the clutch automatically engages the compressor into operation. Therefore, not only forces of inertia of the windmill, i.e. part of its kinetic energy, but also high values of $\bar{M}_{mo}$ at the working rate are profitably used. This method of improving starting quality of a machine is used in certain installations, but it is not very effective in wind machines: in addition to complicating the motor, its reliability is decreased, since clutches usually work unevenly, and dynamic overloads arise during their engagement. For these reasons, piston compressors should also not be used. /145

The compressor is a relatively high-speed machine with which at p = const, practically $M_r \neq f(\omega)$. Combining the compressor with a high-speed windmill, it is possible not to use a reduction gear. With two blades ($Z_n = 7$), moment $\bar{M}_0 = 0.005$, which is 14 times smaller than moment $\bar{M}_n$ (see Fig. 25). Therefore, in order to decrease velocity v_0, it is necessary to increase diameter D_{wm}, but this has a telling effect on the economy of the machine: metal consumption is increased and the coefficient of utilization of established power is decreased. This course is not very acceptable and therefore other methods providing more effective utilization of a pneumatic drive should be sought.

In wind machines, compressed air can be used for power drives of various types, with which there is an intermediate mechanism (air motor, turbine) between the compressor and working machine or directly in the technological process. Thus, air can be a working body accomplishing the lifting of water by means of forcing it out or ejection (Fig. 94).

The use of windmills with blades having channels, along which air is thrown out during rotation due to centrifugal forces and the suction which is used for lifting water is created in the pressure main line, is limited. They can be used for wells having a depth of no more than 7 m. For this reason, air feed into an ejector device is not very promising.

During combination of a wind motor with a rotary compressor and air motor, the maximum pressure in the air main line depends on the frequency of rotation of the windmill. Therefore, the power of the machine changes proportional to ω_{wm}^2 or even to ω_{wm}^3. This eases optimization of the working rates of the drive, especially if two identical machines are used: rotary compressors and air motors driving the pump. The initial moment of resistance in such a compressor is relatively low, but due to the high speed, the use of multistage reduction gears ($i \geq 10$) is necessary. Besides this, it is necessary to install more complex filters and sealing devices on the suction main line to

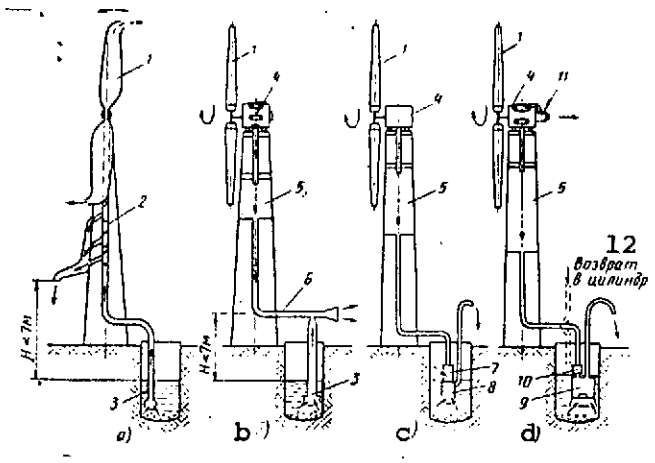

Fig. 94. Layouts of pumping machines with pneumatic drive: a. ejector installation using hollow wings; b. ejector installation with compressor; c. wind machine with rotor compressor and air motor with pump; d. wind compressor machine with displacement pump; 1. windmill; 2. water deflecting device; 3. water intake tube; 4. compressor; 5. receiver; 6. ejector device; 7. air motor; 8. centrifugal pump; 9. pneumatic displacement pump; 10. automatic pump distributor device; 11. compressor relieving device; 12. return to cylinder.

protect the working surface and vanes from wear by abrasive particles in the stream. These compressors are labor consumptive in manufacture, which increases the cost of the machine.

Therefore, the following layout for combination is usually used: a high-speed windmill drives a compressor into rotation, and the compressed air is used for raising water with a pneumatic displacement pump. With this, in some machines, a method for the simple forcing out of water by air is used, and in others, a part of its potential energy is additionally used (the air lift method). In the first case, part of the spent air which still has some residual pressure is released into the atmosphere or flows into the suction main line of the compressor, and in the second, after forcing the water out, the water-air mixture is raised along the pipe and also sent into the reservoir. A singularity of operation of a machine with the displacement pump consists of an intermittent cycle of water feed with a coefficient of efficiency which little depends on the operating rate of the motor. The compressor can work constantly, pumping air into the receiver.

For automation of the operation of drive and pump, devices controlled by a pressure regulator or float are used.

The deficiency in the first layout consists of the fact that a change in some parameters of water feed (H_{Σ}, l_{rt}, d_{rt}) affects the operation of the machine and requires readjustment of the control automatic devices. With a long pneumatic pipeline, the precision of the pump's work is destroyed due to the long time of pressure wave spacing.

Replacing the pressure relay with a float eliminates this deficiency, since the level of water in the pump is the parameter providing the impulse for kicking in the automatic device.

Total time of cycle t_c over which a portion of water equal to the volume of the pump chamber is forced out, is determined by time t_1, over the course of which the receiver is connected to the pump chamber and displacement of the water occurs, and to time t_4, necessary for increasing the pressure in the receiver from p_2 to p_1. The overall coefficient of efficiency of the drive is equal to

$$\eta_{oa} = \eta_p \eta_c \eta_{ln}, \tag{116}$$

where η_p, η_c, η_{ln} are the coefficients of efficiency of the pump, compressor and air lines respectively.

The coefficient of efficiency of the pump is determined by the relationship of the useful work expended on lifting water over time t_c to the isothermal energy of the compressed air:

$$\eta_p = \frac{V_c(p_d - p_0)}{[V_r(p_1 - p_2) + Q_c p_0 t_1] \log \frac{p_1}{p_0}}, \tag{117}$$

where V_c is the volume of a portion of water displaced from the pump over time t_1, in m^3; v_r is the volume of the receiver and air line before the automatic device of the pump; p_0 is the atmospheric pressure; $p_d \approx (0.11 H_\Sigma + 1)$ is the pressure necessary to displace water from the pump, in at; $(p_1 - p_2) \approx (0.2\text{-}0.3)$ is the drop in pressure in the distributor device in at; $Q_c = \frac{(p_d - p_0)Q_p}{p_0 \log \frac{p_1}{p_2} \eta_c}$ is the productivity of the compressor in m^3/sec; /147

$Q_p = V_c/(t_1 + t_4)$ is the productivity of the pump in m^3/sec.

The minimum necessary volume of the receiver is

$$V_{r\ min} = \frac{Q_c p_0 t_4}{p_1 - p_2} \ m^3. \tag{118}$$

If the compressor is connected directly to the pump, the forcing pressure (in n/cm^2) changes from zero up to the pressure corresponding to H_Σ. A machine having a receiver and automatic device begins to work at $p = 0$ only during the initial start. After that, the range of pressure changes is equal to

($p_f > 0$)-($p_c \geq 0.1 H_\Sigma$). The value p_c depends on the depth of source and losses of forcing; p_f depends on the relationship of the volumes of the receiver, air lines and pump, and also on Q_c. With a set Q_p, in accordance with calculations of V. V. Savvotin, the required second's productivity of the compressor is

$$Q_{c.theor.} = \frac{Q_n H_\Sigma t + \frac{F_{ln} H_\Sigma^2}{20}}{10 \log \frac{10 + H_\Sigma}{10}}, \qquad (119)$$

where F_{ln} is the passage section of the water lift pipe in cm^2.

Due to losses in the water lift, a high compressor productivity is needed. These losses are considered with the coefficient $\beta(H_\Sigma, F_{ln}, V_p)$. With H_Σ = 10-20 m, β = 1.3-1.4. For a pump with a chamber having a volume V_p = 15 l, the theoretic specific consumptions of air $q_{a.theo}$ (without consideration of coefficient β are equal to 1.6-2.3 dm^3/l of water for H_Σ = 10-20 m respectively. The actual specific consumptions $q_{a.act}$ are always greater.

Total pressure in m	10	15	20	25	30
η_n	0,56	0,54	0,52	0,51	0,49
$q_{a.act}$ in dm^3/l	2,3—2,8	3,2—3,6	4,8—5,5	6,0—6,5	7,1—7,4

The deficiency in diaphragm and piston compressors is that the power required by them for constant pressure is proportional to the rotation frequency of the windmill, and it is therefore impossible to fully utilize the power of the wind motor during all operating rates and obtain the maximum possible output according to wind conditions.

Knowing the specific consumption of air and the required productivity of the machine, the parameters of the compressor are found. The power required by it is

$$N_c = \frac{Q_c L}{1020 \eta_c' \eta_{is}} \text{ kW}, \qquad (120)$$

where $L = 2.23 \cdot 10^5 p_c \log \frac{p_k}{p_0}$ is the theoretical work of isothermal /148 compression in $n \cdot m/m^3$; p_0 is the suction, usually taken as 0.95 at; p_c is the pressure, which in preliminary calculations can be taken as equal to [(0.11-0.12)H_Σ + 1] at; η_c, η_{is} are the mechanical and indicated isothermal coefficients of efficiency of the compressor.

Knowing the productivity of the compressor and its design parameters, the calculated frequency of rotation is determined, and with set values for Z_n and $v_{n.reg}$, the characteristics of the drive are determined.

$$i = \frac{\omega_c}{\omega_{wm}} = \frac{\pi Q_c D_{wm}}{\tau i_c \lambda V_d Z_n v_{n.reg}}, \qquad (121)$$

where τ is the degree of compression of the compressor; i_c is the number of working spaces (cylinders); λ is the coefficient of feed; V_d is the volume described by the diaphragm during one revolution of the shaft, in dm^3.

Experience shows that, as a rule, the use of reduction gears fastening the windmill onto the compressor shaft should be avoided. For driving the compressor, a windmill is necessary whose minimum diameter will be

$$D_{wm\ min} = \sqrt{\frac{20.4 Q_c L}{v^3_{n/reg} \xi \eta_c \eta_d}}\ m, \qquad (122)$$

where η_d is the coefficient of efficiency of the drive to the compressor.

One of the difficult tasks is providing the beginning of operation of the windmill at $v_{0\ opt}$. Depending on conditions, the starting rates of the compressor are realized at different couterpressures: from $p_n = 0$ up to $p_c = p_{max}$, depending on the adjustment of the automatic device. This determines a different probability of accomplishing the starting of the machine. The most difficult are the starting rates at $p_c = p_{max}$, i.e. when the pressure in the compressor is maximal. These cases arise each time after the receiver is filled with air and before water is forced out of the pump. Having set probability P such that the windmill must stir from its place, the calculated value p_c can be determined. If P is taken to be close to 100%, the calculation must be conducted with $p_{c\ max}$, which requires an increase in the diameter of the windmill.

Due to the large value of $\bar{M}_0$, it is advantageous to use low-speed windmills, since the relationship $\bar{M}_0/\bar{M}_{0\ i=2}$ increases more intensively than the value of $\bar{Z}$ decreases (see Fig. 25). However, in the calculated rate, a reduction in Z_n by 1.85 times (from 5 to 2.7) causes an increase in M_p by only twice. Therefore, /149 the use of low-speed windmills does not provide a large advantage.

Besides this, it is necessary to use reduction gears with high transmission ratios. The combination of water lifts with high-speed windmills and use of devices optimizing the starting and acceleration condition is more effective.

In order to decrease the moment of resistance, it is better to use faster and multicylinder compressors. For a single-stage diaphragm compressor, the moment of resistance depends on the diameter of the diaphragm, the eccentric and the force pressure. The initial moment which must be overcome is

$$M_c = P_d r_c \cos \psi + M_f = (p_c - p_0) F_d r_c \cos \psi + M_f \quad n \cdot m \qquad (123)$$

where P_d is the force acting on the diaphragm, in n; r_c is the radius of the crank (eccentric) in m; ψ is the angle between the axes of the connecting rod and chamber at the moment the pressure valve opens; M_f is the moment of friction in the supports and other components of the drive in n·m; F_d is the usable area of the diaphragm in m^2.

Fulfilling the major condition, we locate $D_{wm\ min}$ or $\overline{M}_0$:

$$\overline{M}_0 \pi \frac{D_{wm}^3}{8} \rho \frac{v_{0\ opt}^2}{2} \geq M_c i. \qquad (124)$$

When using a rotary motion pump, the best layout of a machine with a pneumatic drive is the following: high-speed windmill -- compressor -- pneumatic motor -- pump.

We will examine the possible ways to solve the most important problem of combining a windmill with a compressor -- the improvement of the starting characteristics. This can be achieved by increasing the initial moment of the windmill, using its forces of inertia or, finally, decreasing the moment of resistance of the load during the starting and acceleration period. Other ways (increasing the diameter of the windmill, the number of blades, widening the blades) although simple, are not very effective. A reduction in the calculated values of Z_n or the rotation frequency of the windmill necessitates usage of reduction gears, decreases the coefficient of efficiency and increases the metal consumption of the machine.

Increasing the initial moment of the windmill during starting rates through the use of rotation blades, automatically set at optimum angles is limited and expedient in those cases when the use of a more complex regulation system, with which a mechanism for relieving the compressor is functionally connected, is justified.

For a windmill with $Z_n = 7$, it is possible to increase moment $\bar{M}_0$ by three times by setting the blades at a starting angle of $\phi_s = 31°$ (with $\phi_w = 8°$), although its absolute value is still small ($\bar{M}_0 = 0.016$). Therefore, the beginning of operation can be provided only with high wind velocities (7 m/sec and higher).

Utilization of the forces of inertia of the windmill and its unloading during the starting period are provided by using an automatic clutch (usually centrifugal) in the drive, as well as a sleeve with a pressure sensor which reacts to changes in the velocity of the flow. According to energy-producing indices, this is in large part identical to the unloading of a compressor or an installation of an idle stroke valve on a pump: in the first case, the compressor and pump begin to work when the windmill reaches a certain rotation frequency, and in the second, they work at idle up to definite velocities of the wind. It has been established by th4 calculation method that the reduction in total output of the machine is approximately identical in both cases. We locate the optimal rate for engagement of the sleeve from an equation of dynamics in the acceleration period, considering the adjusted moments of inertia of the windmill J_{wm}, and the parts of the clutch J'_{cl} and transmission J'_{tr} connected with it: /150

$$M_{mo} = (J_{wm} + J'_{cl} + J'_{tr}) \frac{d\omega_{wm}}{dt} \quad n \cdot m \tag{125}$$

If the driving part of the clutch is mounted in the windmill, it is possible to consider only the moment J_{wm}, since $J'_{cl} << J_{wm}$. Upon engagement of the clutch, we have

$$M_{nw} + M_p = (J_{wm} + J'_{cl} + J'_{tr} + J'_c) \frac{d\omega_{wm}}{dt} = \Sigma J'_{\Sigma} \frac{d\omega_{wm}}{dt} . \tag{126}$$

If the size of M_{mo} of the windmill is insufficient to overcome the moment M_r, then part of the kinetic energy is also used

$$\underline{E}_J = \left(\frac{\omega^2_{cl_1} - \omega^2_{cl_2}}{2} \right) \Sigma J \quad J, \tag{127}$$

where ω_{cl_1}; ω_{cl_2} are the angular velocity of the clutch and windmill at the moment of clutch engagement and its disengagement or during the period when the equation $M_{mo} = M_r$ is established.

For the wind velocity found to be $v_{0\ opt}$, the value ω_{cl_1} can be determined according to combined characteristics. Engagement

occurs at $M_{mo} > M_r$. In our example for the "Vikhr'" machine (Fig. 95), $n_{1\ opt} \approx 100$ rpm at $v_{0\ opt} = 4$ m/sec, and $p_2 = 2$ at. The moving moment transmitted to the compressor is created by aerodynamic forces on the windmill, and is also formed due to utilization of part of the kinetic energy E_J of the windmill:

$$E_J = (M_r - M_{mo})\ \omega_{wm.av}. \tag{128}$$

From expression (127), we locate the maximum value

$$\omega_{cl_1} = \sqrt{\frac{2}{cl_2} + 2\frac{E_J}{\Sigma J}} \tag{129}$$

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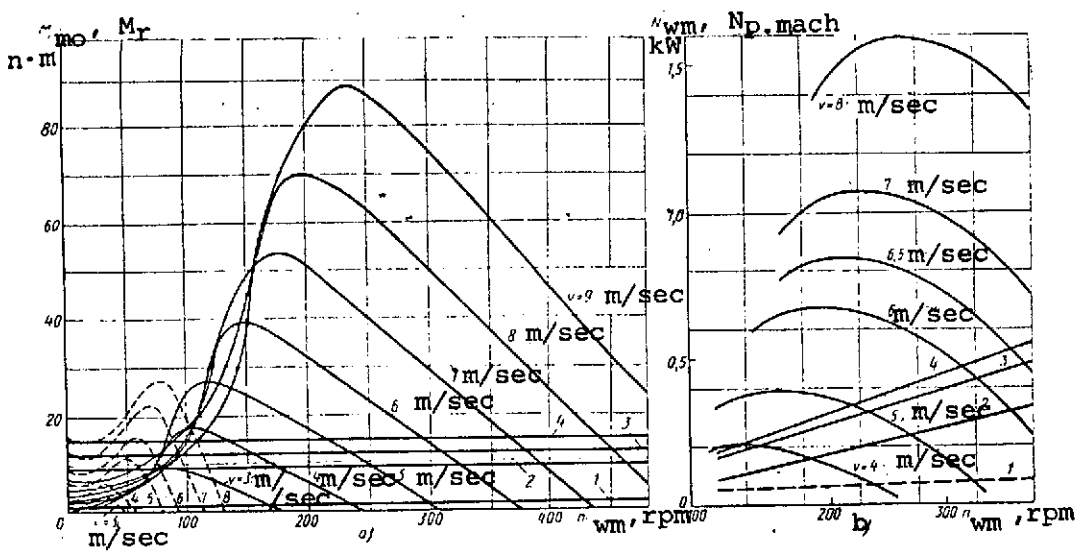


Fig. 95. Combined moment (a) and working (b) characteristic of the windmill $D_{wm} = 4$ m and compressor of the "Vikhr'" machine: 1. M_r at $p_{exc} = 0$; 2. Same, with $p_{exc} = 1$ at; 3. Same with $p_{exc} = 2$ at; 4. Same with $p_{exc} = 3$ at; solid lines $\phi = 8°$; dotted lines $\phi = 31°$.

In order to decrease the number and frequency of clutch engagements, it is better to engage it at ω_{wm} somewhat higher than ω_{cl1}. From Fig. 95 it is evident that if pressure $p = p_w = 3$ at in the pressure main line of the compressor (in the receiver) then with $\phi_0 = \phi_w = 8°$, $v_{0\ min} = 10.5$ m/sec, and with $\phi_0 = \phi_{opt}$, /152

$v_{0\ min} \approx$ 7 m/sec. Such a machine is not very effective. It must also be considered that the maximum (momentary) values M_{or} of the compressor are higher than the average values of the moment; this deteriorates conditions of starting.

A situation in which the windmill begins to rotate but does not achieve the calculated rate as a result of intensively increasing resistance can take place. With this, the gradient of M_r will be greater than the gradient of M_{mo}. For an analysis of these phenomena, it is possible to make use of the graphical-analytical method of computing the acceleration characteristics of a machine with changing values of the moment (Table 18, Fig. 96).

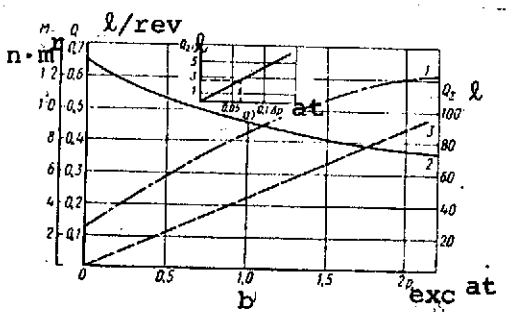


Fig. 96. Auxiliary graphs for locating acceleration characteristics of a pneumatic wind machine: a. increase in pressure per 1 revolution of windmill; b. change in value of M_r, Q and Q_Σ at various counter-pressures; 1. $M_{r.av}$; 2. Compressor feed per 1 shaft revolution; 3. total quantity of compressed air.

If in all initial rates the start of the machine without counter-pressure is secured by connecting the consumer to the pressure main line when $M_{mo} > M_r$, the values of v_0 and $v_{work\ min}$ are reduced. In this way, one of the difficult problems connected with utilization of a wide range of wind velocities and an increase in economy of the machine is solved. In our case, if p_c = 1 at, the windmill begins to work at $v_0 \approx$ 3.5 m/sec and at approximately these same velocities at optimal rates can overcome a load resistance for p_c = 3 at.

The method of improving characteristics of machine with pneumatic drive provides relieving of the compressor during the starting and acceleration periods, in combination with rotating the blades to optimal starting angles in cases where the blades are controlled by a centrifugal regulator. /153

Unloading in dependence on rotation frequency is accomplished using a by-pass device on the compressor, which is controlled by the windmill regulator (see Fig. 41), with a special load-bearing regulator or with an electrical system (electromagnetic valve). The simplest and most advantageous is the first device; the use of electrical systems is less economic, since a special generator is required, complicating the machine and increasing power losses.

The compressor, at a low frequency of windmill rotation, releases air into the atmosphere, and when rotation frequency increases to the extent that moment M_{mo} becomes sufficient to

TABLE 18. FOR COMPUTATION OF ACCELERATION CHARACTERISTICS OF A PNEUMATIC MACHINE

t in sec	$z = \frac{\omega_{c+1}}{\frac{v}{R}} = \frac{R\omega_{c+1}}{v}$	$M_{mo} = f(Z)$ (conducted according to characteristic)	$M_{mo} = M_{mo} R^3 \frac{v^2}{2} = 15.7\, M_{mo} v^2$ in n·m	M_r in n·m (according to Fig. 96)	$M_\Sigma = M_{mo} + M_r$ in n·m	$\omega = \frac{M_\Sigma}{J_{wm}}$ in rad/sec^2	$\omega_{av} = \omega_c + \omega \frac{\Delta t}{2}$ in rad/sec	$\omega_{c+1} = \omega_c + \omega\Delta t$ in rad/sec	Number of revolutions over time Δt $n_\Sigma = \frac{\Delta t}{2\pi} \omega_{av}$	Q in l/rev (according to Fig. 96)	Q_Σ clo $= n_\Sigma Q$ in l	Δp in at (according to Fig. 96)	$p_{c+1} = p_c + \Delta p$ in at	$p_{av} = p_c + \frac{\Delta p}{2}$ in at	$n_{c+1} = 9.57\omega_{c+1}$ in rpm	$n_{av} = 9.57\omega_{av}$ in rpm
								in m/sec								
5	0,03	0,022	4,25	2,5	1,75	0,025	0,1127	0,1752	0,09	0,65	0,058	0,001	0,001	0,0006	1,67	1,08
10	0,1	0,022	4,25	2,5	1,75	0,025	0,237	0,3	0,189	0,65	0,123	0,003	0,004	0,0025	2,87	2,27
15	0,17	0,022	4,25	2,5	1,75	0,025	0.3	0,425	0.24	0,65	0,156	0,004	0,008	0,006	4,06	2,87
25	0,243	0,022	4,25	2,5	1,75	0,025	0.55	0,675	0,88	0,64	0,564	0,012	0,02	0,014	4,46	5,26
35	0,385	0,0225	4,27	2,5	1,77	0.025	0,8	0,925	1.28	0,64	0,82	0,018	0,038	0,029	8,85	7,65
45	0,53	0,023	4,4	2,7	1,7	0.024	1.045	1,165	1 68	0,635	1,07	0,023	0,061	0,005	11,15	10,0
55	0,665	0,0235	4,5	2,9	1,6	0,023	1,28	1,395	2.05	0,63	1,29	0,029	0,09	0,076	13,4	12,3
65	0,98	0,027	5,2	3,1	2,1	0.03	1.54	1,695	2,17	0.62	1,53	0,034	0,124	0,107	16,3	14,7
75	0,97	0,028	5,3	3,3	2,0	0,028	1,83	1,975	2,94	0,61	1,79	0,041	0,165	0,144	18,9	17,5
85	1,13	0,033	6,37	3,6	2,77	0,35	2.132	2,325	3.42	0,6	2,06	0,045	0,21	0,187	22,3	20,4
95	1,33	0,041	7,9	3,9	4,0	0.057	2.61	2,895	4,18	0.59	2,46	0,055	0,265	0,238	27,7	25,0
105	1,65	0,056	10,8	4,3	6,5	0,093	3.36	3,22	5.37	0,56	3,0	0,067	0,332	0,298	36,5	32,2
115	2,18	0,076	14,6	4,7	9,9	0,141	4.52	4,23	7.2	0,55	3,96	0,088	0,42	0,342	50,0	43,2
								in m/sec								
2	0,04	0,022	5,5	2,5	3,0	0.043	0,043	0,086	0.0274	0,65	0,0178	0,001	0,001	0,0005	0,0083	0,4
4	0,043	0,022	5,5	2,5	3,0	0.043	0.129	0,172	0,041	0,65	0,027	0,001	0,002	0,0015	1,65	1,2
10	0,086	0,022	5,5	2,5	3,0	0,043	0,301	0,43	0.29	0.65	0,19	0,004	0,006	0,004	4,2	2,95
15	0,215	0,025	5,5	2,6	2,9	0.041	0.55	0,67	0,262	0,64	0,168	0,004	0,01	0,008	6,4	5,3
20	0,335	0,022	5.5	2,6	2,9	0.041	0,76	0,85	0,605	0,64	0,37	0,008	0,018	0,014	8,3	7,4
25	0,425	0,0225	5,6	2,6	3,0	0.043	0.957	1,065	0.762	0,64	0,46	0,01	0,028	0,023	10,0	9,2
30	0,53	0,023	5,8	2,7	3,1	0.0443	1.18	1,287	0.94	0,635	0,6	0,014	0,042	0,035	12,5	11,3
35	0,643	0,0235	5,9	2,75	3,15	0.045	1,51	1,547	1,2	0,635	0,76	0,017	0,059	0,044	16,8	14,8
40	0,87	0,027	6,8	2,9	3,9	0,056	2,017	1,8	1,73	0,63	1,09	0,025	0,084	0,072	22,0	19,4
45	1,15	0,033	8,3	3,0	5,3	0,076	2.5	2,1	2,16	0,625	1,35	0,03	0,114	0,099	25,9	23,9
50	1,35	0,04	10,0	3,2	6,8	0.1	2.95	2,4	2.56	0,615	1,58	0,036	0,15	0,132	30,7	28,2
55	1,6	0,049	12,3	3,5	8,8	0.126	3.51	2,8	2 82	0,605	1,71	0,038	0,188	0,169	36,6	33,5
60	1,91	0,065	16,25	3,7	12,55	0,18	4,28	3,3	3.43	0,595	2,18	0,05	0,238	0,213	45,3	41,0
65	2,36	0,087	21,8	4,1	17,7	0.253	5.36	4	4,2	0,58	2,45	0,055	0,293	0,266	57,5	51,5
70	3,0	0,12	30,0	4,5	25,5	0.366	6,91	4,7	5,53	0,57	3,15	0,07	0,363	0,301	75,0	66,0

overcome the calculated load, the by-pass device automatically closes and air flows into the receiver, which has a check valve. Therefore, with a decrease in wind speed, air compressed earlier does not escape from the receiver into the atmosphere. This method of unloading is still one of the most effective, also because it provides a functional relationship between the four parameters -- load, windmill rotation frequency, wind velocity and position of the by-pass device valve.

Unloading in dependence on the wind velocity is accomplished using a device controlled by the wind regulator. This method of unloading, used on the "Lastochka" machine (see Fig. 41), is convenient to use in motors which are regulated by the windmill's turning out of the wind using a lateral plane, and using the latter as the wind regulator. The work of this regulator is determined primarily by the velocity of the flow; the effect of a change in windmill rotation frequency is relatively small. Consequently, the relationship between the action of the unloading mechanism and the operating rates of the machine is unclear, and herein lies the deficiency of this method. In order to eliminate the effect of micropulsation of the flow caused by its turbulence on the work of the wind regulator, angular position locators for the side plane empennage are used.

The by-pass of air from the pressure main line of the compressor into the atmosphere is stopped when velocity v = 4-4.5 m/sec and the moment $M_{mo} > M_r$. However, at a velocity of v = = 5-7 m/sec, a case can occur in which, due to an intensive increase in air pressure in the receiver, the machine cannot rapidly attain its calculated rate, especially if in the initial period there is a considerable counterpressure in the receiver. The probability of such cases can be reduced to a minimum if a check valve is not installed in the receiver, although this will also cause some losses of air.

Unloading the compressor according to air counterpressure in the pressure main line provides utilization of the counterpressure as a parameter determining the rate of air feed into the atmosphere or to the consumer. The cylinders of the compressor are not /156
connected to the receiver directly, but through an intermediate pressure relay or regulating valve. Using the increase in resistance in the passage sections of the valve (relay) during the increase in the compressor's productivity, caused by the increase in the windmill's rotation frequency, it is possible to close the valve and direct air into the receiver. When the windmill's rotation frequency and productivity decrease, the valve disconnects the compressor cylinders from the receiver and connects them with the atmosphere.

By comparison with normal machines, the output of machines equipped with relieving systems is increased by 3-5 times, which is caused by utilization of the most frequently repetitious wind

velocities. An additional optimization of starting rates using rotating blades still further increases the economy of the machine. The reduction in total productivity of the machine by comparison with the theoretically possible (maximum) according to wind conditions due to bypassing part of the air into the atmosphere at low wind velocities, or due to the windmill working with the clutch at idle (for the same range of wind velocities) is approximately identical and relatively small.

This method of unloading provides the possibility for having a wind motor with sort of changing speed, using in it the advantages of a low-speed machine during the starting rates (high values of moment $\bar{M}_0$) and those of a high-speed motor at the main working rates (higher amount for values ω_{wm} and $\bar{M}_{mo}$ at Z_n).

With a set pump feed Q_t, we find the calculated productivity of the compressor Q_c starting from the fact that $0.1H_\Sigma q_a$ volumes of air, adjusted for normal conditions are required per unit of water volume raised with a pressure of H_Σ. If $\omega_c = \omega_{wm}$, then, having taken the number of cylinders as equal to i_{cy}, we determine the theoretical volume of a cylinder V_{cy} in cm^3, corresponding to a single stroke of the piston (diaphragm), by using the expression

$$Q_c = \frac{V_{cy} i_{cy} n_c k_w \lambda}{1000} \quad m^3/min \qquad (130)$$

where k_w is the coefficient considering the decrease in cylinder volume (by comparison with the theoretical) as a result of deformation of the diaphragm.

Known V_{cy}, it is not difficult to determine the required cylinder diameter d_{cy} and piston stroke h.

For smoother operation of the windmill and easing of conditions of its acceleration, multicylindered compressors are better. For design considerations and conditions of operation, it is not expedient to have more than four cylinders due to their lesser effect on equalization of forces (moments) and an increase in the number of valve groups. By comparison with a single-cylinder one, during utilization of a four-cylinder compressor, unevenness of the moment of resistance on the shaft is decreased by almost six times (Fig. 97). The addition of two more cylinders only increases the evenness by 8%.

Knowing the power required by the compressor and the average moment $M_{wm.av} = 9.5 \cdot 10^3 (N_c/n_c)$, calculated rotation frequency of the compressor is refined according to combined characteristics and its design parameters are corrected. /157

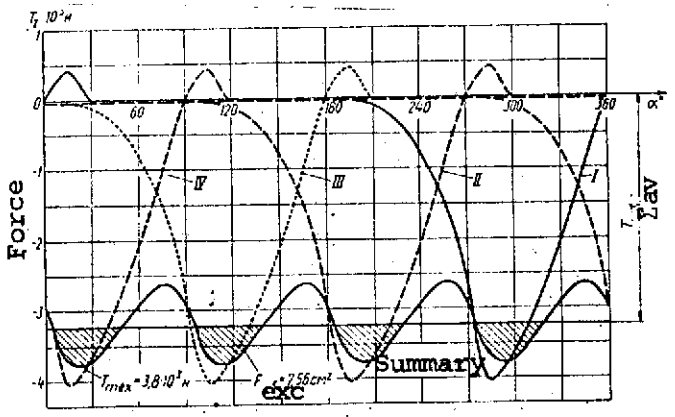


Fig. 97. Changes in summary tangential forces T_Σ in a four-cylinder diaphragm compressor: I-IV. number of diaphragm blocks of cylinders.

Investigations have shown that an unloading device connected with a centrifugal windmill regulator (see Fig. 41) works reliably. It is desirable to engage it at a windmill rotation frequency which is 15-20% in excess of the frequency at which the starting spring of the regulator is fully compressed, and it is better to reduce the value of velocity v_c, /158 according to which the calculated productivity is determined, by 1-2 m/sec by comparison with $v_{n.reg}$. This increases provision of working rates for the machine (Table 19) and the coefficient of utilization of the established power.

In the normal starting rate, a machine with unloading begins to work at $v_0 \approx 4$ m/sec, while without unloading it works at v_0 = 5.5-6.3 m/sec; without optimization of starting, velocity v_0 increases by approximately 2.5 times.

TABLE 19. STARTING RATES OF "VIKHR'" MACHINE.

Starting Conditions	Angle φ°	Init. pressure in Receiver in at			
		1	2	3	3.8
		v_0 in msec			
Relieving device engaged	34	3,6	4,0	4,1	3,9
Same	7	4,0	—	—	—
Relieving device disengaged	34	3,65	5,55	6,3	5,15
Same	7	6,7	9,9	10,35	9,8

§ 3. Hydraulic Drive in Wind Machines

The use of the hydraulic drive can provide maintenance of consistency in the operating rates of a machine during a changing frequency of windmill rotation through automatic reduction of the

transmission functions of the drive; reduction of dynamic overloads and metal consumption of the unit; utililization of grouped drive; and a relatively high value for the coefficient of efficiency. At the same time, due to the complexity of the machine, an increase in cost and increased requirements for servicing, it can be expected that hydraulic drive will be used only in medium-power machines.

A hydrodynamic transmission is not very adaptable for wind machines, since its elements (centrifugal pump, radial-axial hydraulic turbine) work at high velocities, and their coefficient of efficiency drops sharply with a change in rotation frequency. Displacement (hydrostatic) drives consist of lower-speed machines and have a stable coefficient of efficiency, and their use is therefore preferable for wind motors.

We will examine three basic types of hydraulic drives.

1. Fluid, which is the working body, sets the hydraulic motor, which is connected with a load or pump pulsator into motion. For circulation along the closed cycle (the so-called closed system), a non-freezing mixture can be used, simplifying operation of the hydraulic drive in the wintertime. In the pumping machine, it is also possible to accomplish the take-off of water from the main flow directed to the consumer. In this case, periodic topping up of the system with water is not required.

A machine in which pumps and motors of the screw type are used is more effective. The pump, on the shaft of which is fastened a high-speed windmill, feeds fluid into the hydraulic engine (motor), which is connected to the water lifting pump (Fig. 98). With this layout, one can expect a high value for the coefficient of efficiency of the machine (η_{oa} = 0.4-0.5), and a relatively stable rotation frequency in a wide range. The use of diaphragm pumps in a wind machine with hydraulic drive for high pressure heads is less promising in the coming years. These pumps are clumsy, have a low coefficient of efficiency, require use of a piston pulsator and are inconvenient in operation.

Multicylinder axial-piston pumps have been successfully /159
tested in a machine with a wind motor. Compact hydraulic reversing motor-pumps, having a high coefficient of efficiency (0.8-0.85), a weight and relatively even loading moment have been developed up to the present. Using these motor-pumps (one in the hydraulic drive system, the other for lifting water), it is possible to obtain η_{oa} = 0.5-0.6, approximating machines with piston pumps in this parameter.

2. The liquid (water), partially extracted from a common reservoir, is fed into an inertia pump ("Don" type), in which, under the pressure of the water, vibration of the valve occurs, providing alternate opening of the valve groups and lifting of water.

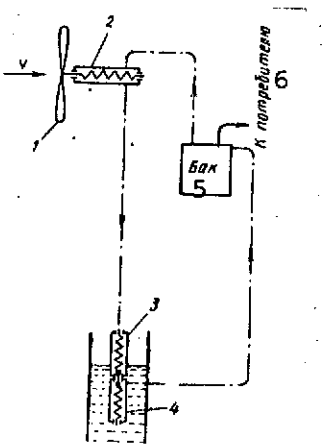


Fig. 98. Layout of hydraulic drive with screw-type pump: 1. high-speed windmill; 2. screw hydraulic pump; 3. screw hydraulic motor; 4. screw-type water-raising pump; 5. tank; 6. to consumer.

A screw pump is best used as the hydraulic pump creating pressure in the working liquid.

The coefficient of efficiency in this machine is expected to be within limits of 0.4-0.45. However, the use of windmills with regulators to provide great evenness in rotation frequency and devices in the drive system to stabilize the feed of liquid is required. The necessity of this is caused by the fact that the "Don" is a resonance type pump.

3. Water, partially extracted from the pressure main line, is used as the working liquid, forced by the pump into a water stream apparatus, which raises water to the surface. This layout provides high reliability of the machine, although the coefficient of efficiency, especially with high pressure heads, will not exceed 20%. Therefore, the area of wise utilization of these machines should apparently be limited to pressures of under 25 m.

The optimum parameters of a hydraulic system and machine can be obtained with variant calculations. However, when basing some initial data and combining, it must be kept in mind that although the displacement types pumps used have a moment which depends little on the rotation frequency of the windmill, with an increase in rotation frequency, losses in the hydraulic drive and in the pressure pipeline increase (the working pressure of the pump changes), proportional to the square of the velocity ($v_{1n}^2/2g$) of the flow of the liquid in the pipes. This will lead to an increase in the moment of resistance of the drive and to better loading of the machine at high wind velocities. /160

For optimization of starting rates, it is also necessary to use the same devices as in machines with pneumatic drive, or the unloading valves used in the machine with a screw-type pump.

§ 4. Combination of a Wind Motor and Working Machine Using Electric Drive

An electric drive in wind machines, which consists of a generator, electric motor, working machine, power wires, and also devices for regulating voltage, protecting the generator and motor from too-high settings and, in some cases, of switching apparatus,

not only provides the transmission of power, but simultaneously allows the establishment of a rotation frequency of the working organs which is necessary for effective operation of the machine. The advantages of electrical drive over drives of other types are the following: operational flexibility, less metal consumption, the possibility for installation of a motor at a distance from the consumer or well, where the wind rate is better, and utilization of the machine for providing power to a group of consumers, which makes it universal.

At the same time, its use in wind machines increases the necessary minimal working velocities of the wind, requires the use of reduction gears and more highly improved systems of regulation, and decreases the overall coefficient of efficiency, especially at low wind velocities. When working with a centrifugal pump, the economy of a machine having a power of up to 2 kW with electrical drive is lower than when using mechanical or pneumatic drives, and therefore its use is most rational in areas with relatively high average wind velocities (4.5 m/sec and higher), and in cases when other types of machines are less effective.

The functioning elements of the drive in a wind machine have a number of singularities which must be considered when examining its operating rates and selecting parameters. These singularities include power by a generator of a motor having comparable power [$N_{mo} = (0.6\text{-}0.8)N_{gen}$], which makes the starting of the motor more difficult; significant changes in frequency, and also in voltage at the terminals of the generator, and in a number of cases, great length of the current-conducting lines.

Electric drive provides for use of a high-speed windmill. Therefore, the use of direct current generators having a high initial moment of resistance and requiring more exacting care is undesirable (with the exception of low-power charging devices. However, a three-phase alternating current system consisting of a synchronized generator and short-circuited asynchronized motor is used. The selection of the type of generator, system of excitation and law of regulation of voltage $U(\omega,N_n)$ to a significant degree determines the effectiveness of the machine and the rate of its operation [27, 46]. In the majority of cases, the law for regulating voltage is determined by the fact that maintenance of $U/f \approx$ const is strived for, although for some types of loads (for instance, active), it is better to maintain constancy of voltage within a wider range of current frequency. /161

The use of asynchronized generators with self-excitation from static condensers simplifies the construction of the machine and its operation and reduces overloads. An asynchronized generator is, as a rule, cheaper than a synchronized one of equal power. In recent years, the possibility for using asynchronized generators has broadened due to the use of variconds -- nonlinear capacitances (controlled condensers). However, the capacitance of condensers,

which are necessary for stable operation of a generator-motor system is relatively large, which increases the cost of the machine. Therefore, in many cases, synchronized generators with self-excitation from solid rectifiers or generators of three-phase current with hook-shaped poles on the rotor, semiconductor rectifiers and a contact-free system of voltage regulation are preferable.

In specific conditions of the use of an independent wind machine, the effectiveness of an electrical drive increases when an increased (by comparison with industrial (50 Hz)) current frequency is used, since the relative weight and dimensions of the electrical equipment are decreased and additional possibilities appear for increasing the power of the asynchronized motor, which is power by a generator of comparable power.

In pumping machines with electrical drive, pumps are required which are calculated for pressure heads of 1.5-2 times greater than those which are determined by the depth of the water source, and which work at significant deviations from the nominal rotation frequency (up to $0.5n_n$). The use of a high-frequency drive eases the problem of creating these pumps. At the same time, great difficulties arise in providing high reliability for equipment (generators, motors, pumps, and so forth) which are still cheaper than those serially produced.

The problem of stable starting of the electric motor by the wind generator can be solved in three ways: 1) by connecting the motor to an excited generator which is working at idle; 2) frequency starting up the motor, previously connected to the terminals of the generator, which begins to work with the excitation circuit closed; 3) closing the excitation circuit of the generator, with the motor connected to its terminals. Investigations in this country and abroad showed that electrical pumping wind machines acting with a frequency and voltage with a constant value can work automatically, without special automation devices, which simplify /162
the construction and servicing of the installation and increases its effectiveness. It is proposed that, in particular, the pump motor be connected directly to the generator, so that they stop (at $v = v_0$) and start (with $v \geq v_0$) together. With this, the concept of nominal (normal values for current frequency and voltage lose their meaning, since the machine works with a changing rotation frequency right up to the limiting values, limited by the mechanical strength of the components, parts and electrical part of the generator and motor ($v = v_{work\ max}$). Then the notion of combining a motor with a pump will consist of the fact that the aerodynamic and loading characteristics are as close together as possible, but that the pump, upon a decrease in wind velocity, does not cause a transfer of the machine to unattainable low rotation frequency rates, when its rotary moment is sharply reduced.

The use of vibration pumps with electromagnetic drive in a machine with a high-speed windmill eliminates a number of deficiencies present in inertia water lifts with mechanical drive: dynamic overloads and metal consumption are reduced, and output is increased in regions with an average annual wind velocity of $\bar{v}_a > 5$ m/sec.

The principal possibility for effective combination of pumps of this type with high-speed windmills is brought about by the fact that during operations at rates with changing values for current frequency and voltage at the generator terminals, the pump loads the motor with a power which changes as a function of current frequency and voltage. The vibrator valve has an amplitude of lift and a number of motions per minute which also depends on these parameters, and can in essence fulfill the task of a stroke as a result of the conformities indicated. /163

With the aforestated plan for a magnetoelectric pump system and selection of a law for regulating generator voltage, it is possible to attain the best combination of working characteristics of the motor and loading curve of the pump.

With constant values of voltage and frequency of current feed, the moment of resistance of the pump is almost independent of pressure, and productivity falls with an increase in pressure. With a reduction in frequency and a constant generator voltage, the moment of resistance and the productivity of the pump are decreased almost proportionately, but with a simultaneous reduction in generator voltage the curve of moment M_r is close to a parabola, which provides the possibility of working at optimal rates in a wide range of wind velocities.

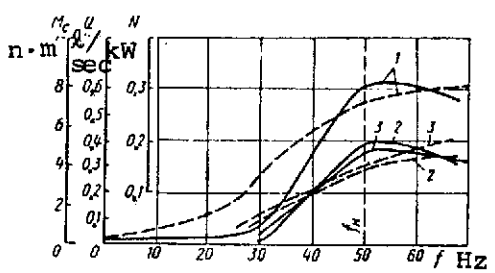


Fig. 99. Characteristics of model of wind machine with vibration pump (GAB-0.5 generator; H_Σ = 10 m): 1. moment; 2. productivity; 3. power (solid lines -- work according to normal schematic; broken lines -- on negative half-cycle).

One of the singularities of a pump with electromagnetic drive is the large value of the inductive component of resistance X_i by comparison with the active component R_p [X_i = = (8-9)R_p], which has a great effect on the load and work of the generator with a normal pump connection schematic. The system is loaded with a large reactive power and it is therefore necessary to increase the power of the generator significantly by comparison with the nominal power of the pump. An attempt to utilize double half-cycle rectification in the schematic did not yield a

desirable effect, since the constant component appearing in the transfer process destroys the normal operation of the pump.

Experimental investigations of a model with single-phase GAB-0.5 and GAB-1 generators, having a power of 0.5 and 1 kW showed that the pump acts stably in a wide range of current frequency and voltage values. When working on the negative half-cycle, M_r changes proportional to ω_{wm}, while at the same time the resistance moment and required power for a normal schematic increase, and then decline as frequency f increases (Fig. 99).

Chapter 8. Aerodynamic Calculation of a Wind Machine

For many motors, requirements set forth by the load to the windmill can be fulfilled with a varying number of blades, i.e. with various speed Z_n. Therefore, for selecting a windmill corresponding to the conditions presented, the most fruitful is the course of variant calculations.

Initial data when selecting a windmill are results of computations on optimization, which give values for the base velocities of the wind ($v_{n.reg}$ and v_0) and also the most advantageous windmill diameter. If the calculated power $N_{load\ w}$ required by the working machine is set, then, setting an approximate value for the overall coefficient of efficiency of the wind motor, the power which the windmill must develop is found:

$$N_{wm} = N_{load\ w}\eta_{mech}\eta_{el} \quad kW \tag{131}$$

where η_{mech} is the mechanical coefficient of efficiency of the drive from the windmill to the working machine; η_{el} is the coefficient of efficiency of electrical machines (generator, electric motor).

The aerodynamic layout of the windmill is selected depending on the speed of the working machine and its initial moment of resistance. The characteristics of airfoil profiles are taken from special reference books. The "Espero," "Gettingen-399" and "Inversiya ellipsa" profiles are most frequency used for high-speed motors.

Having determined the calculated power of the windmill according to formula (131), its diameter is computed

$$D_{wm} = 28.4\sqrt{\frac{N_{wm.w}}{\pi\rho v_{n.reg}^3\xi}} \quad m. \tag{132}$$

The number of modules of Z_n is given or, if the calculated number of revolutions is known, determined according to the formula

$$Z_n = \frac{\omega_n R}{v} = \frac{\pi n_c R}{30v},$$

from which

$$n_c = \frac{30Z_n v}{\pi R}. \tag{133}$$

Thickness of the blade profile is selected from the basis of /165 data from wind tunnel models with consideration for design aspects, chief of which is provision of the required blade strength. Usually, the thickness of the profile δ at the tip of the blade is taken to be within limits of (0.1-0.15) of the width b of the blade. Thickness δ can remain constant along the entire length of the airfoil or, for instance, for windmills with rotating blades, can increase up to (0.2-0.3)b as it approaches the hub.

In order to solve equations determining the form and number of blades, and also the coefficient of utilization of the wind energy ξ, several values for the coefficient of braking e are substituted, and then the values for the ideal coefficient ξ_i is determined according to formula (65). Using equation (65) and (76), the values for actual coefficient ξ are determined, and then, constructing a graph of $\xi(e)$, ξ_{max} and the values of e corresponding to it are located. The results of calculation are conveniently presented in the form of Table 20. With this, it is necessary to have the auxiliary diagram of values $C_y(\alpha)$ and $\mu(\alpha)$ shown in Fig. 100. At the end of calculations, the angle of blade inclination is determined in the calculation sections taken, which are located at a relative radius $\bar{r} = r/R$, where r is the distance to the section from the axis of rotation of the windmill.

Substituting a different number of blades i_b, blade width b is found and a final value for coefficient C_y is refined according to it.

$$b = \frac{(\overline{i_b b} C_y)}{i_b C_y} R; \tag{134}$$

$$C_y = \frac{(\overline{i_b b} C_y)}{i_b b}, \tag{135}$$

where $\overline{i_b b} = (i_b b)/R$ is the relative total width of the blade.

Also conducting calculations for all blade sections selected from $\bar{r} = 1$ up to $\bar{r} = 0.2$, the data necessary for planning the windmill and its characteristics are obtained (Table 21).

Since the blade profile is thicker close to the hub, then, so that the blade thickness does not become extremely high, it is necessary to use high values of C_y for sections remaining further from the blade tip. One usually tries to set the width of the /167 blade at $\bar{r} = 0.2$ at 1.3-1.7 times higher than at the tip and a line connecting the ends (tails) of the section would be straight.

Aerodynamic computations for a windmill require some practice. In particular, frequently after the calculations are completed,

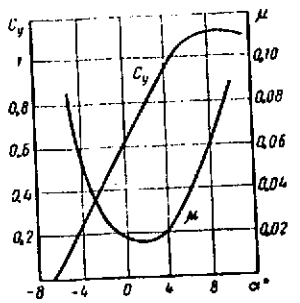


Fig. 100. Curves for determining coefficients $C_y(\alpha)$ and $\mu(\alpha)$.

it is necessary to correct the angles of inclination ϕ, striving so that the products, i.e. the lines connecting the similar points on the airfoil in different sections of the blade are straight. This simplifies the technology of manufacturing the blade, and makes it more acceptable with respect to esthetics.

If as the result of design development and strength calculations it is expected that the thickness of the profile is insufficient for placing load-bearing elements of the airfoil in its contour (stringers, root, etc.), the calculation is repeated, beginning with determination of a value $\sqrt{1 + Z_u^2}$ (Table 21), increasing the profile thickness during this.

The characteristics of a windmill, i.e. the dependencies $M_{mo}(Z)$ and $\xi(Z)$, are first computed for each section of the blade (see Table 20) and the graphs of $\Delta M_{mo}(Z)$ and $\Delta M_{mo}(\bar{r})$ shown in

TABLE 20. EXAMPLE OF WINDMILL CALCULATION (D_{wm} = 3.6 m; Z_n = 3.5; i_b = 4; PROFILE -- "INVERSIYA ELLIPSA;" RELATIVE THICKNESS 0.2)

$\bar{r}$	$\xi_i = \lambda\frac{1-\bar{r}}{1+\bar{r}}$	$\bar{r}_f \approx \frac{\bar{r}}{1-\bar{r}}\left\{\frac{8\sqrt{1+\left(\frac{1-\bar{r}}{Z_n}\right)^2}}{(1+\bar{r})\,i_b\bar{b}\,Z_n} - \frac{1}{\sqrt{1+\left[\frac{i_b\bar{b}\,Z_n}{\pi(1-\frac{\bar{r}}{\bar{r}})}\right]^2}}\right\}$	$\bar{r}_p \approx 2\mu'\left[\frac{1-\bar{r}}{Z_n} + \frac{Z_n}{3(1-\bar{r})}\right]$	$\bar{r}_m \cong \frac{\xi_i}{2}\cdot\frac{\eta_i^2}{Z_n^2}\ln\frac{R}{r_0}$	$\xi = \xi_i\left[1-\left(\frac{r_0}{R}\right)^2 - \bar{r}_f - \bar{r}_p - \bar{r}_m\right]$
0,3	0,646	0,113	0,135	0,158	0,476
0,35	0,674	0,075	0,079	0,085	0,478
0,4	0,685	0,034	0,036	0,036	0,465

$Z = Z_n\frac{r}{R}$	$Z_u \approx \frac{Z}{2(1-\bar{r})}\left[1+\sqrt{1+\frac{\xi_i}{Z^2}}\right]$	$\bar{b}^{bC_y} = \frac{8\pi\frac{r}{R}\cdot\frac{\bar{r}}{(1+\bar{r})(1-\bar{r})^2}}{(Z_u+\mu)\sqrt{1+Z_u^2}}$	$b = \frac{\left(\bar{b}^{bC_y}\right)}{i_b\bar{b}^{bC_y}}R$	$\varphi = \operatorname{arcctg} Z_u - \alpha$

$\frac{r}{R}$	Z	$\frac{Z}{2(1-\bar{r})}$	$1-\frac{\sqrt{1+\frac{\xi_i}{Z^2}}}{\ldots}$	Z_u	$8\pi\frac{r}{R}$	$Z_u+\mu$	$\frac{\bar{r}}{(1-\bar{r})(1-\bar{r})^2}$	$\sqrt{1+Z_u^2}$	$\bar{b}^{bC_y}$	C_y	μ	b	α	$\beta = \operatorname{arcctg} Z_u$	$\varphi = \beta - \alpha$
1,0	3,5	1,3	2,026	5,40	25,08	5,48	0,614	5,55	0,508	0,654	0,016	0,35	0° 30'	10° 23'	9° 53'
0,8	2,8	1,3	2,042	4,40	20,10	4,42	0,614	4,52	0,618	0,696	0,016	0,40	0° 54'	12° 48'	11° 54'
0,6	2,1	1,3	2,072	3,35	15,08	3,37	0,614	3,50	0,788	0,788	0,017	0,45	2° 06'	16° 39'	14° 33'
0,4	1,4	1,3	2,160	2,33	10,06	2,35	0,614	2,53	1,046	0,942	0,025	0,50	3° 55'	23° 20'	19° 25'
0,2	0,7	1,3	2,540	1,37	5,02	1,39	0,614	1,70	1,304	1,066	0,047	0,55	6° 08'	36° 08'	30° 00'

Note. Here $\mu' = \frac{C_x}{C_y}$ is the average value of the inverse lift-to-drag ratio of the airfoil profile along its entire length; $Z = \frac{\omega R}{v}$ is the number of moduluses or speed; $Z_u = \cot\beta = \frac{\omega r + u_1}{v - v_1}$ is number of relative moduluses; $i_b b$ is the summary width of the blades; $\overline{i_b b} = \frac{i_b b}{R}$ is the relative summary with of the blades.

TABLE 21. EXAMPLE OF CHARACTERISTICS CALCULATION FOR A WINDMILL
(D_{wm} = 3.6 m; 1_b = 4; Z_n = 3)

$\bar{r}$	$\varphi°$	$\frac{b}{r}$	$\alpha°$	$\beta = \varphi + \alpha$	$Z_u = \frac{c}{r}\lg(\varphi+\alpha)$	μ	$\frac{e}{(1+e)(1-e)^2} = C_y \frac{(Z_u+\mu)\sqrt{1+Z_u^2}}{\frac{8\pi\bar{r}}{b}}$					
							C_y	$\frac{8\pi\bar{r}}{\frac{b}{r}}$	$Z_u+\mu$	$\sqrt{1+Z_u^2}$	$\frac{e}{(1+e)(1-e)^2}$	e
1,0	11,2	0,67	−5	6,2	9,21	0,109	0,150	37,70	9,32	9,26	0,343	0,24
			−4	7,2	7,91	0,058	0,250		7,97	7,96	0,422	0,25
			−2	9,2	6,17	0,029	0,440		6,20	6,25	0,482	0,29
			+2	13,2	4,26	0,016	0,780		4,42	4,38	0,401	0,25
			6	17,2	3,23	0,037	1,040		3,28	3,39	0,313	0,28
			8	19,2	2,87	0,058	0,110		2,93	3,04	0,262	0,20
0,8	14,1	0,74	−6	8,1	7,03	0,292	0,056	27,20	7,33	7,10	0,107	0,09
			−4	10,1	5,61	0,058	0,250		5,67	5,70	0,296	0,23
			−2	12,1	4,67	0,029	0,440		4,70	4,77	0,363	0,25
			+2	16,1	3,46	0,016	0,780		3,48	3,60	0,359	0,25
			6	20,1	2,73	0,037	1,060		2,77	2,91	0,315	0,29
			8	22,1	2,47	0,058	1,110		2,53	2,66	0,275	0,20
0,6	18,3	0,81	−6	12,3	4,59	0,292	0,056	18,64	4,68	4,68	0,068	0,0
			−4	14,3	3,92	0,058	0,250		3,97	4,04	0,215	0,17
			−2	16,3	3,42	0,029	0,440		3,44	3,56	0,288	0,21
			+2	20,3	2,70	0,016	0,780		2,72	2,88	0,329	0,25
			6	24,3	2,21	0,037	1,080		2,25	2,42	0,311	0,25
			10	28,3	1,85	0,089	1,120		1,94	2,11	0,246	0,16
0,4	23,2	0,89	−6	17,2	3,23	0,292	0,056	11,30	2,53	3,38	0,059	0,0
			−4	19,2	2,87	0,058	0,250		2,93	3,04	0,197	0,1
			−2	21,2	2,58	0,029	0,440		2,61	2,77	0,281	0,2
			+2	25,2	2,12	0,016	0,780		2,14	2,35	0,347	0,2
			6	29,2	1,79	0,037	0,060		1,83	2,05	0,352	0,2
			10	33,2	1,53	0,089	1,120		1,62	1,83	0,295	0,2
0,2	33,9	0,89	−6	27,9	1,89	0,292	0,056	5,66	2,19	2,14	0,046	0,0
			−4	29,9	1,74	0,058	0,250		1,80	2,01	0,160	0,1
			−2	31,9	1,61	0,029	0,440		1,64	1,90	0,242	0,1
			+2	35,9	1,38	0,016	0,780		1,40	1,70	0,329	0,2
			6	39,9	1,20	0,037	1,060		1,23	1,56	0,360	0,2
			10	43,9	1,04	0,089	1,120		1,13	1,44	0,323	0,2

$\bar{r}$	$\frac{1+e}{e}$	$1-\mu Z_u$	$Z_u+\mu$	$\Delta\bar{M}_{m0}$	$Z_u(1-e)$	$\frac{1}{Z_u\frac{1+e}{e}}$	Z	$Z_{III} = \frac{Z}{\bar{r}}$
8,00	5,12	−0,004	9,32	−0,0006	6,970	0,021	6,950	6,95
	4,59	+0,540	7,97	+0,1184	5,700	0,028	5,670	5,67
	4,42	0,821	6,20	0,2400	4,360	0,037	4,320	4,32
	4,70	0,932	4,42	0,3590	3,110	0,050	3,060	3,06
	5,40	0,880	3,28	0,3980	2,400	0,057	2,440	2,44
	5,98	0,833	2,93	0,3810	2,290	0,058	2,230	2,23
5,12	11,44	−1,050	7,33	−0,0640	6,370	0,012	6,360	7,94
	5,55	+0,674	5,67	+0,1100	4,380	0,032	4,350	5,43
	4,95	0,865	4,70	0,1916	3,485	0,043	3,440	4,30
	4,98	0,945	3,48	0,2800	2,590	0,058	2,530	4,30
	5,34	0,899	2,77	0,3110	2,100	0,069	2,030	3,16
	5,79	0,857	2,53	0,3000	1,954	0,070	1,884	2,36
2,88	16,60	−0,340	4,88	+0,0121	4,300	0,013	4,290	7,15
	6,81	+0,773	3,97	0,0823	3,240	0,037	3,200	5,33
	5,63	0,901	3,44	0,1340	2,685	0,052	2,633	4,38
	5,21	0,957	2,72	0,1940	2,064	0,071	1,993	3,32
	5,40	0,918	2,25	0,2170	1,706	0,084	1,622	2,70
	6,23	0,835	1,94	0,1990	1,500	0,086	1,414	2,36
1,28	18,9	+0,057	3,53	+0,0011	3,050	0,016	3,030	7,57
	7,25	0,833	2,93	0,0503	2,415	0,048	2,367	5,92
	5,70	0,925	2,61	0,0795	2,030	0,068	1,962	4,90
	5,09	0,966	2,14	0,1140	1,605	0,092	1,513	3,78
	5,04	0,934	1,83	0,1300	1,347	0,111	1,236	3,09
	5,57	0,864	1,62	0,1230	1,195	0,117	1,078	2,69
0,32	23,50	+0,448	2,19	+0,0028	1,800	0,023	1,777	8,88
	8,36	0,899	1,80	0,0192	1,504	0,069	1,435	7,18
	6,30	0,953	1,64	0,0295	1,305	0,099	1,206	6,03
	5,23	0,978	1,40	0,0426	1,054	0,138	0,916	4,58
	4,96	0,956	1,23	0,0501	0,895	0,168	0,727	3,63
	5,28	0,907	1,13	0,0487	0,796	0,182	0,614	3,07

Figs. 101 and 102 are constructed. The relative specific moment $\Delta\bar{M}_{mo}$ of a section attributable to the blade element with a length Δr and determined according to the equation

$$\Delta\bar{M}_{mo} = 8\,\frac{e}{1+e}\cdot\frac{1-\mu Z_u}{Z_u+\mu}\,\bar{r}^2\Delta r,$$

depends on the speed of the section and the radius at which it is located.

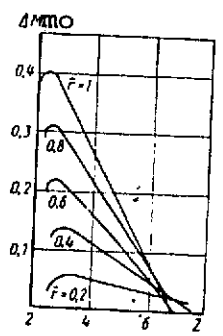


Fig. 101. Change in relative rotary moment along blade sections of a windmill: $i_b = 4$; $Z_n = 3$.

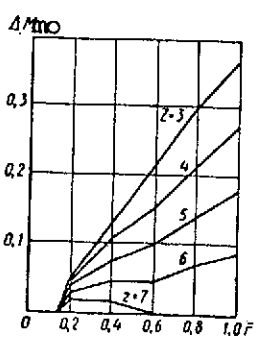


Fig. 102. Characteristic of relative rotary moment along a blade for a windmill: $i_b = 4$; $Z_n = 3$.

The graph in Fig. 102 is an approximate solution of the equation /170

$$\bar{M}_{mo} = 8\,\frac{e}{1+e}\int_{\bar{r}_0}^{\bar{R}}\frac{1-\mu Z_u}{Z_u+\mu}\,\bar{r}^3\,d\bar{r}. \qquad (136)$$

This indicates that the area limited by the axis of the abscissa and the lines of the moments is equal to the integral, taken within limits of from $\bar{r} = 0.2$ to $\bar{r} = 1$. Replacing this area with an equal-sized triangle with a base of $\bar{r} = 1$, the value for the moment, numerically equal (in the scale of the ordinate taken) to the height of the triangle is located.

The aerodynamic characteristic of the windmill (Fig. 103) is obtained by projecting the height of the triangles taken according to Fig. 101 for each value of Z along the ordinate axis and number of moduluses along the abscissa axis. In order to consider so-called tip losses which arise as the result of the fact that the blade has a terminal length and reduce the moments of the windmill, the concept of the moment of tip losses is introduced

$$M_{t.l} = \frac{N_{t.l}}{\omega_{wm}},$$

where $N_{t.l}$ is the power of tip losses.

The relative value of $\bar{M}_{t.l}$ is determined according to the equation

$$\bar{M}_{t.l} = \frac{4\varepsilon^2}{z_n(1+\varepsilon)} \times \left\{ \frac{\sqrt{1+\left(\frac{1-\varepsilon}{z_n}\right)^2}}{(1+\varepsilon)\frac{z}{b_n}} - \frac{1}{\sqrt{1+\left[\frac{\frac{z}{b_n}}{\pi\left(1-\frac{\varepsilon}{2}\right)}\right]^2}} \right\}. \quad (137)$$

Then the actual characteristic $\bar{M}_{mo}(Z)$ is obtained, subtracting the value of $\bar{M}_{t.l}$ from $\bar{M}_{mo}$, which was calculated according to formula (136). Multiplying $\bar{M}_{mo}$ by Z, the curve $\xi(Z)$, which has received the name power curve is located.

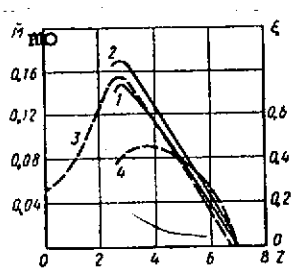


Fig. 103. Aerodynamic characteristic of windmill (i_b = 4; Z_n = 3).

1 - Theoretical;
2 - Theoretical without considering tip losses;
3 - Test; 4 - $\xi(Z)$

Chapter 9. Systems for Automatic Rotation Frequency Regulation and Power Limitation /171

Due to constant changes in the velocity of the wind and, consequently, energy of the flow within wide limits, the power of a wind motor changes. Thus, with an increase in velocity by three times, the energy of the flow increases by 27 times. The power which a windmill can develop increases correspondingly. In order to protect the motor from overloads and limit the rotation frequency of the windmill within set limits, systems for automatic regulation are used. Using them, with a wind velocity which exceeds calculated, the power of the motor is limited, not allowing it to grow higher than the calculated power, determined by conditions of strength, and the necessary evenness of rotation of the working machines connected is provided. The regulator is usually also an element of the starting and stopping mechanism.

Regardless of the method, the main principle of regulation comes down, in the final analysis, to a change in the lift force on the blades and the moment of aerodynamic forces on the windmill. Selection of one system of regulation or another is determined by the requirement set forth by the working machine, the diagram for utilizing the machine, its speed, design of the windmill and other conditions.

Two basic methods of regulation are used: changing the position of the entire windmill in the stream (directing it out of the wind) and turning the blades (or their end portions) by the proper angles of attack. Methods of regulation using butt-end valves, slats (air brakes) and other devices installed on the airfoils for the purpose of changing their lift force, and also inclining the blade at some angle from the plane of rotation of the windmill

(so-called umbrella regulation) are also known. However, these methods are used relatively rarely, since they do no provide high quality regulation and reliability.

Regulation by directing the windmill out of the wind is usually used in low-speed machines, and more rarely in machines of medium speed or windmills with three or four rigidly fastened blades. With this method, aerodynamic forces are used.

In systems of regulation by turning the blades, which are used in high-speed motors, the following forces are used: centrifugal (regulating elements and the mass of the blade itself), aerodynamic or both simultaneously. The basic stages of designing any regulator provide calculation of its kinematics, static and dynamic characteristics, evaluation of the quality of regulation and development of the regulator's design.

§ 1. Calculation and Design of Regulators for Low-Speed Wind Machines /172

Regulation through directing the windmill away from the wind is accomplished due to the fact that at flow velocities higher than calculated, the plane of rotation of the wheel is rotated relative to the vector of the velocity and with a flow velocity greater than the maximum working one it is located in the plane of action of the stream. Therefore, the moment of aerodynamic forces on the windmill becomes equal to zero and the machine stops.

Two types of mechanisms with which this principle of regulation is realized are known.

In mechanisms of the first type (Fig. 104, a) the moment of aerodynamic forces on the windmill arising as the result of the shifting of its axis of rotation by the amount ε relative to vertical axis z is used for turning the wheel relative to axis z. This shift has received the name eccentric. Regulation occurs in the following manner: with wind velocities less than calculated ones, the plane of rotation of the windmill is perpendicular to the direction of the wind, and the power which the motor develops is the maximum. In this position I, the windmill is held by the force of the preliminary tension of the regulation spring 2, in tension between the tail and the regulator mechanism, which is connected to the head. /173 With an increase in flow velocity, frontal pressure on the windmill increases, moment $M_{zz} = P_{frt}\varepsilon$ increases and the windmill rotates by angle γ_1 (position II), and therefore the power is almost unchanged. With very high wind velocities, the windmill rotates by 90° and moment M_{mo} becomes equal to zero (position III). The moment from tangential forces T, acting on the plane of the windmill (arm l_T) also participates in regulation, and in the presence of a vertical shaft, the moment of reaction

forces $M_{r.w}$ also participates. Usually, it is directed toward the same side as the moment of forces of resistance, which imparts to the regulation system an additional protective property against accidental disengagement of the load: in this case, the wheel deflects, i.e. turning away from the wind occurs at lower velocities v.

The mechanism is used in low-power machines, in which there is not a vertical shaft and reaction forces therefore do not arise. This system of regulation is not highly accurate at a low rotation frequency, considerable overloads and gyroscopic moments can arise in the mechanism, and the range ($v_{n.reg}$ - $v_{work\ max}$) is smaller than in other systems of regulation. The advantage of the mechanism is its simplicity and reliability of operation.

In mechanisms of the second type, forces arising on the lateral plane (surface), located on the windmill are used for moving the windmill away from the wind (Fig. 104, b).

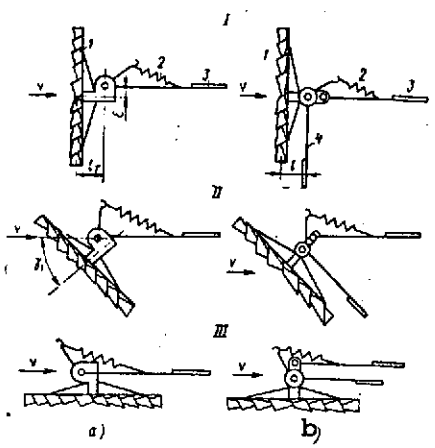


Fig. 104. Diagram of regulating power and rotation frequency of a windmill by moving it out of the wind: a. due to an eccentric; b. using a lateral plane; I-III. windmill positions; 1. windmill; 2. spring; 3. tail; 4. lateral plane.

The regulation process with this occurs more smoothly, the sensitivity of the mechanism is increased, and the angular velocities of head rotation are reduced. The mechanism is usually used in motors with a vertical shaft, and also in cases where a higher speed windmill is directed out of the wind. Naturally, the lateral plane increases metal consumption.

The advantage of regulation by moving the windmill out of the wind is the presence of aerodynamic damping in the system, which eases construction of regulators which are dynamically stable at all working rates, including those in which the rotation frequency of the windmill is reduced during an increase in wind velocity. This occurs within a wide range of changes for value ω_{wm} and v, right up to $\omega_{wm} = 0$ with $v = v_{max}$. In this way, the property for automatic self-protection of the windmill during storms is present.

The system is highly flexible, it is not difficult to adjust for various operating rates, and during calculation, the required conformity for regulation

of parameters is comparatively easily reached, since the basic active forces participating in the regulation process, including part of the forces of resistance, are aerodynamic and can be changed in the necessary directions.

During planning and calculation of these regulation systems, it must be kept in mind that the moment of aerodynamic forces on the empennage of an isolated tail $\bar{M}_{ta}$ significantly exceed (by 2-4 times) the moment $\bar{M}_{ta}$ of a tail located on a working windmill. The change in working rates of a windmill only slightly affects the value of forces acting on the lateral plane, but the value of relative moments of aerodynamic forces arising on the lateral plane $\bar{M}_{pl}$ in the presence of the windmill can differ by 20-40% from the values of $\bar{M}_{pl}$ obtained during wind tunnel tests of an isolated plane. The effect of the shape of the plane is relatively small. The characteristic of moments of the aerodynamic forces rotating the windmill relative to its vertical axis $\bar{M}_{rot}(\gamma_1)$ must inasmuch as possible respond to conditions in which in the initial section $\bar{M}_{wm}$ and the derivative $\partial\bar{M}_{wm}/\partial\gamma_1$ are greater than zero. This is promoted by an increase in tangential forces and a decrease in the values for the moments of frontal pressure forces acting in a direction opposite to the action of the moment of tangential forces. The latter can be achieved by shifting the center of curvature toward the trailing edge of the windmill. /174

The order for calculating parameters of regulation is the following. For maintenance of nominal power at wind velocities exceeding the calculated one, it is necessary to rotate the windmill by an angle determined in orientation according to the expression $\cos\gamma_1 = v_w/v$. At v_w = 8 m/sec

v in m/sec	8	9	10	15	20	25	30
γ_1 in degrees	0	20	36	57	66	71	75

The mechanical system examined has three degrees of freedom. The conditions of its static equilibrium are:

$$M_{mo} - M_r = 0; \quad M_{rot} - M_{sp} = 0; \quad M_{ta} + M_{sp} = 0, \qquad (138)$$

where M_{sp} is the moment created by the spring of the regulation mechanism.

The equation contains nonlinear functions $\bar{M}_{mo}$, B, $\bar{T}$ and $\bar{Z}(Z,\gamma_1)$, $\bar{M}_{ta}(Z,\gamma_1,\gamma_2)$ and $\bar{M}_{pl}(Z,\gamma_1)$ determined by experimental means (here γ_2 is the angle of rotation of the tail) [35, 37]. The values of some coefficients are given in Table 22. /175

TABLE 22. COEFFICIENTS $\bar{Y}$ AND $\bar{X}$ OF FORCES ACTING ON A LOW-SPEED WINDMILL ACTING ALONG THE RESPECTIVE AXES

Coefficient	Angle γ_1 in degrees							
	0	10	20	30	40	50	60	70
$\bar{X}$	0,8	0,78	0,72	0,60	0,45	0,34	0,26	0,22
$\bar{Y}$	0	0,04	0,10	0,15	0,20	0,20	0,24	0,30
$\bar{a}=\frac{a}{R}$	0	−0,005	−0,01	0,015	0,05	0,06	0,10	0,12

Here a is the displacement of the point of incidence of force X relative to the axis of rotation of the windmill.

We will examine a mechanism built according to the diagram in Fig. 105, a. If the axes of rotation of the tail and head do not coincide, i.e. $\varepsilon_{ta} \neq 0$, the forces of the regulation mechanism spring, as the result of its lenghtening by the amount Δl_{sp} with changes in angle γ_1 and γ_2 will be

$$P_{sp} = C(a + \Delta l_{sp}) = C[a + \sqrt{\rho^2 + l^2 - 2\rho l \cos[(\beta + \nu) - (\gamma_1 - \gamma_2)]} - \sqrt{\rho^2 + l^2 - 2\rho l \cos[(\beta + \nu) - \gamma_{2_0}]}],$$

where C is the stiffness of the spring; a is the preliminary tension of the spring in its initial position ($\gamma_1 = 0$; $\gamma_2 \neq 0$); ν is the angle between the axis of the windmill and a line connecting the axis of rotation of the tail and head; γ_{2_0} is the angle between the velocity vector of the airflow and the plane of the tail at the initial moment of regulation. /176

$$\rho = \sqrt{r^2 + \rho_{ta}^2 - 2r\rho_{ta}\cos(\theta - \nu)};$$
$$\beta = \arcsin\left[\frac{r}{\rho}\sin(\theta - \nu)\right];\quad \rho_{ta} = \frac{e_{xa}}{\sin\nu}.$$

Knowing the velocity at which the windmill must be totally directed out of the wind, we obtain

$$\bar{M}_{mo.Z} = 0(\gamma_1\ st)\ R^3\rho\ \frac{v_{st}^2}{2} = M_{0c}; \qquad (139)$$

$$\bar{M}_{rot.Z} = 0(\gamma_1\ st)\pi R^3\rho\ \frac{v_{st}^2}{2} = M_{sp}(\gamma_1\ st,\ \gamma_2). \qquad (140)$$

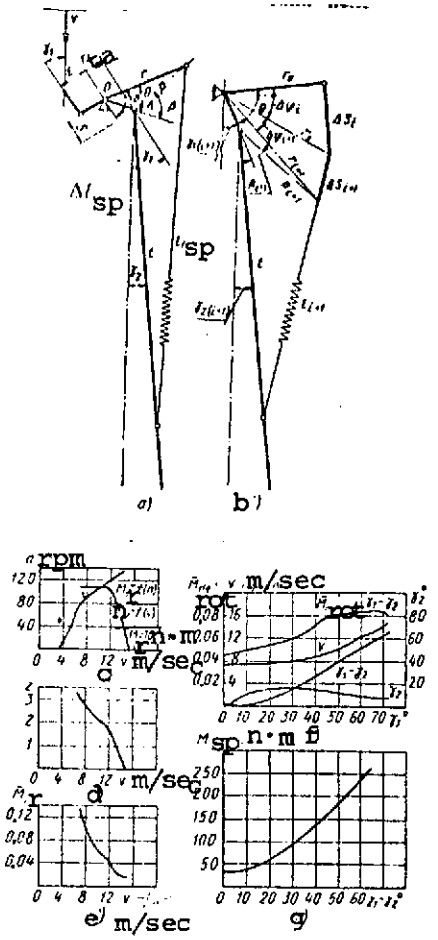


Fig. 105. Calculating a regulation system: a. diagram of mechanisms; b-g. characteristics of elements of the system.

Calculations are completed for a series of spring mechanisms with different values for θ, C, and r. If a multi-linked lever or formed element (so-called "snail") is used in the mechanism instead of a simple lever, it is possible to obtain any set characteristic for moments created by the spring and rotating the windmill relative to its vertical axis.

Example of calculation. Given a motor, in which $D_{wm} = 4$ m; $i_b = 6$; $Z_n = 2.8$; $v_w = 7$ m/sec; $v_0 = 4$ m/sec; $v_{work\ max} = 15$ m/sec; gear ratio of the upper reduction gear $i_{ur} = 2.15$; and the reach of the plane of rotation of the windmill from the axis of the head $l_{wm} = 0.24$ m. The windmill has the following values for moments: nominal $\bar{M}_{mo} = 0.16$, initial $\bar{M}_0 = 0.06$ (see Fig. 93); calculated moment of load resistance $\bar{M}_r = 33$ n·m; and the relative reaction moment at the calculated rate $r_w = 0.136$.

We compute the dimensions of the laterial plane, beginning with the fact that in the initial position (v_w) moment M_{rot} is greater than moment M_r. Then,

$$M_{rot.c} = \frac{2M_{rot.c}}{\pi R^3 \rho v_w^2} = \frac{2 \cdot 33}{3.14 \cdot 2^3 \cdot 1.23 \cdot 7^2} = 0.043$$

(where $\rho = 1.23$ kg/m^3 is the density of air), and the necessary coefficient for the moment of the lateral plane is

$$\bar{M}_{pl\ 0} = \bar{M}_{rot.c} + \frac{\bar{M}_{rw}}{i_{ur}} = 0.043 + \frac{0.136}{2.15} = 0.106.$$

The rotational moment of the head is

$$\bar{M}_{rot} = B\bar{Z} + 0.12\bar{T} + \bar{M}_{pl} - 4.65\bar{M}_{mo} \qquad (141)$$

Beginning with characteristics for empennages of various shapes [37], we determine

$$\bar{F}_{pl}\bar{l}_{pl} = \frac{\bar{M}_{pl\,0}}{k_{pl}C_{R_{60°}}} = \frac{0.106}{1.2\cdot 0.91\cdot 1.1} = 0.0885,$$

where k_{pl} is the coefficient of lateral plane, equal to 1.2; $C_{R60°}$ is the coefficient of resultant forces on lateral plane at the angle of its initial setting $\alpha = 60°$ ($C_{R60°} = 0.91C_{R90°} = 0.91\cdot 1.1$).

Taking $\bar{l}_{pl} = 1.25$, i.e. $l_{pl} = 1.25\cdot 2 = 2.5$ m, we obtain $\bar{F}_{pl} = \frac{0.0885}{1.25} = 0.0705$; $F_{pl} = 0.0705\pi R^2 = 0.705\cdot 3.14\cdot 2^2 = 0.885$ m^2. For a plane with a square empennage $a_{pl} = \sqrt{0.885} = 0.94$ m.

In the initial position the moment of the tail $\bar{M}_{ta.c} = \bar{M}_{in.c} = 0.043$. With an empennage bent along a circular arc, the coefficient of loading necessary for obtaining the set moment of the tail is /177

$$C_R = \frac{\bar{M}_{ta\,0}}{\bar{F}_{ta}\bar{l}_{ta}k_{ta}} = \frac{\pi\bar{M}_{ta}\lambda_{ta}}{\bar{h}^2_{ta}\bar{l}_{ta}k_{ta}} = \frac{3.14\cdot 0.043\cdot 1.2}{0.575^2\cdot 1.5\cdot 0.69} = 0.47.$$

Taking $l_{ta} = 1.5$, the coefficient of the tail $k_{ta} = 0.69$, and the height of the empennage $\bar{h}_{ta} = \frac{h_{ta}}{R} = 0.575$, we obtain $\bar{F}_{ta} = \frac{\bar{h}^2_{ta}}{\pi\lambda_{ta}} = \frac{0.575^2}{3.14\cdot 1.2} = 0.087$. Then $F_{ta} = 0.087\pi R^2 = 0.087\cdot 3.14\cdot 2^2 = 1.09$ m^2 and the chord $b_{ta} = \frac{F_{ta}}{h_{ta}} = \frac{1.09}{0.575\cdot 2} = 0.95$ m.

The calculation of moments of the tail at various angles of γ_1 and γ_2 is conducted according to the methodology described in Chapter 10.

A spring regulation mechanism must provide obtainment of a set characteristic of $n(v)$ at the maximum motor load (Fig. 105, b). In order to calculate the mechanism, it is necessary to construct the auxiliary graphs (Fig. 105, c, d, and e), recommended by M. O. Frankfurt. Using them, the basic dependency $M_{sp}(\gamma_1 - \gamma_2)$ necessary for designing the mechanism (Fig. 105, e) is obtained. With $\rho_{ta} = 0.128$ m, t - 2 m, length r - 0.3 m of the first link,

placed at an angle of 110° to the axis of the tail and a so-called central tail (ε_{ta} = 0 and ν = 0), we obtain

$$\rho = 0.282 \quad \text{and} \quad \beta = 84°.$$

The initial force of the spring at γ_2 = 2°30' is equal to approximately 120 n.

For the spring rigidity taken C = 1.0 km/m and a $\approx$ 0.12 m, we obtain the parameters of the regulation mechansim with three degrees of levers (Table 23), using the formula

$$M_{sp} = P_{sp} l \sin\chi = Cl\left[a + \sqrt{\rho^2 + l^2 - 2\rho l\cos[(\rho+\nu)+(\gamma_1-\gamma_2)]} - \sqrt{\rho^2 + l^2 - 2\rho l\cos[(\beta+\nu)-\gamma_{2_0}]} \times \sin\operatorname{arctg}\frac{\rho\sin[(\beta+\nu)+(\gamma_1-\gamma_2)]}{l-\rho\cos[(\beta+\nu)+(\gamma_1-\gamma_2)]}\right], \tag{142}$$

After calculating the static characteristics, it is necessary to investigate the dynamic properties of the system. A wind motor, as a system with three degrees of freedom, is described by equations which after all transformation and simplifications have the following appearance [37]:

$$\left.\begin{aligned} J_0\frac{d\omega}{dt} &= M_{mo} - M_r \\ J_1'\frac{d^2\gamma_1}{dt^2} + \varkappa_1\frac{d\gamma_1}{dt} + J_s\frac{d^2\gamma_2}{dt^2} &= M_{rot} - \frac{M_r}{i_w} - M_{sp} \\ J_2\frac{d^2\gamma_2}{dt^2} + \varkappa_2\frac{d\gamma_2}{dt} + J_s\frac{d^2\gamma_1}{dt^2} &= -M_{ta} + M_{sp} \end{aligned}\right\} \tag{143}$$

where $\varkappa_1$, $\varkappa_2$ are the coefficient of aerodynamic damping; $J_0 = J_{0x} + J_{s_y} i^2_{l.w}$ is the adjusted moment of inertia of the windmill and links (parts) kinematically connected with it; $i_{l.w}$ is the gear ratio between the windmill and link; $J'_1 = J_1 + m_2 S^2$ is the moments of inertia of the head and the entire system relative to axis y (where $J_1 = J_{1y} + J_{sy}$); m_2 is the mass of the tail; S is the distance between the axes of rotation of the head and tail; J_{1y} is the moment of inertia of the rotating head with the windmill relative to axis y; J_{sy} is the adjusted moment of inertia of the tail relative to the axis of rotation of the head; J_2 is the moment of inertia of the head relative to the axis of its hinge. /178

We will conduct an analysis of the equation according to common criteria used in the theory of regulation. We note that in the majority of wind motors, the axes of rotation of the tail and head do not coincide, and this leads to the action of additional forces on the tail. Coincidence of the axes reduces inertial forces which, in turn, promote improvement of the operation of the regulation system. With displaced axes, an additional moment of inertial forces M_{in} is directed toward the side opposite that of M_{pl} (Fig. 106), and is composed of moments of the reciprocally moving and turning masses.

$$M_{in.rec}^{max} = \frac{m_{ta}}{2} \omega_0 r^2; \tag{144}$$

$$M_{in.trn}^{max} = \frac{m_{ta}}{2} \omega_0 r l_{cg}; \tag{145}$$

$$M_{in\ \Sigma} = \frac{m_{ta} \omega_0 r}{2} (r + l_{cg}). \tag{146}$$

TABLE 23. PARAMETERS OF REGULATION MECHANISM

No. of stages	l_i, m	ρ_i, m	γ_i, °	$\Delta\omega_i$	$\sum \Delta\omega_i$	ΔS_i, m	$\sum \Delta S_i$, m	$\gamma_i - \gamma_1$, °	γ_1, °	γ_2, °	$\gamma_{\Sigma i}$, °
1	0,3	0,363	129	—	—	—	—	10	10	0	10
2	0,43	0,324	41,5	80	80	0,478	0,478	23	38	15	28
3	0,636	0,507	2,5	28	108	0,324	0,324	65	72	7	31

Note. Angles γ_1 and γ_2 are taken as positive during calculation for counterclockwise direction.

Thus, for the "Lastochka" machine, the moment of inertial forces from the mass of the tail alone reaches 20 n·m. It is comparable with the moment created by the lateral plane (30-33 n·m). When designing a mechanism with a central axis of tail rotation, special attention must be paid to reduction of the forces of friction in the bearings. For this, it is necessary to reduce the weight and span of the tail, and introduce roller bearings. The moment of friction is equal to /179

$$M_{r.ta} = \frac{2G_{ta} l_{cg}}{l_{br}} f_{fr} d_{br} \tag{147}$$

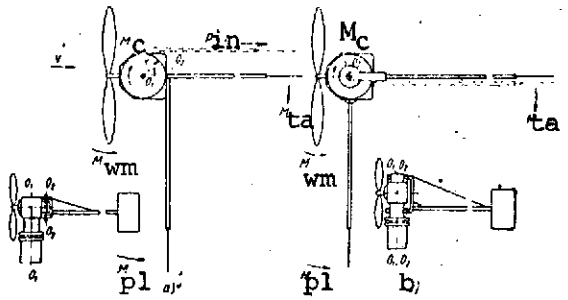


Fig. 106. Determination of inertial regulation forces with removed (a) and central (b) axes of tail rotation: O_1O_1 -- axis of rotation of the head; O_2O_2 -- axis of rotation of the tail.

where G_{ta} is the weight of the tail and its empennage; l_{cg} is the distance of the center of gravity of the tail with its empennage relative to the axis of its rotation; f_{fr} is the coefficient of friction in the bearings; d_{br} is the diameter of tail supports (bearings); l_{br} is the distance betweeen the bearings.

In order to eliminate the harmful effect of the forces of friction in the pivot sleeve and its weight during the regulation process, it is recommended that a device is introduced, with which the sleeve is automatically fixed in the upper position and does not move during the regulator's operation. In cases in which there is no fixing device, the weight of the sleeve must be overcome by the force of the preliminary tension of the regulator spring, which deteriorates the action of the regulator. The moment of /180
resistance $M_{r.s}$, which is created by the sleeve and the parts connected with it, with angles between the axis of the tail and plane equal to 50-60° is comparable with the moment of the spring. We will write the equation for the adjusted forces and moments toward the axis of the main spring (Fig. 107), which, after transformation, takes on the following appearance:

$$M_{r.s}=\left\{(G_s+G_{cb})\left[1+\left(\frac{d_s}{h_s}+\frac{d_r}{d_s}\right)f_{fr}\right]\right\}l_1\cos\alpha, \qquad (148)$$

where G_s, G_{cb} are the weight of the sleeve, cable and parts connected with them; d_s is the internal diameter of the sleeve; h_s is the height of the sleeve; d_r, d_s are the diameter of rollers and their shafts respectively; f_{fr} is the coefficient of friction.

§ 2. Basic Principles of Designing Regulation Systems for High-Speed Wind Machines

Depending on the forces used during regulation, the following basic systems are differentiated (Fig. 108, a-e):

The stabilizer regulation system (the system of G. Kh. Sabinin and N. V. Krasovskiy), when the blade or its tip rotates

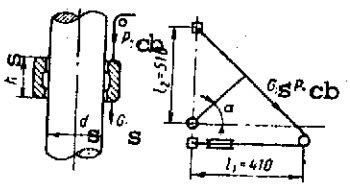


Fig. 107. Calculating forces of resistance in elements of the starting-stopping mechanism.

due to the effect of aerodynamic forces arising on the stabilizer, which is controlled by a centrifugal regulator.

This system of regulation acts in the following manner (Fig. 108 a and Fig. 109). The tip portion of blade 4 with stabilizer 1 (Fig. 108, a) can freely rotate relative to the axis of spar 2. Due to the stabilizer, the end of the blade, similar to a wind vane, is set in a direction relative to the flow W at angle of attack α, determined by the angle of stabilizer setting with respect to the blade (Fig. 109, position I). When the machine is started, spring 5 (Fig. 108, a) of the regulator is freed and the stabilizer therefore rotates at some angle. The lifting force acting on it creates moment M_{st} and rotates the blade, on which the forces setting the windmill into rotation begin to act (Fig. 109, position II). If as the result of a change in the windmill's rotation frequency and wind velocity angle α is changed, moment M_{st} acquires another value and the blade turns so that angle α is regained. The setting work of the windmill before the beginning of regulation is characterized by position III in Fig. 109.

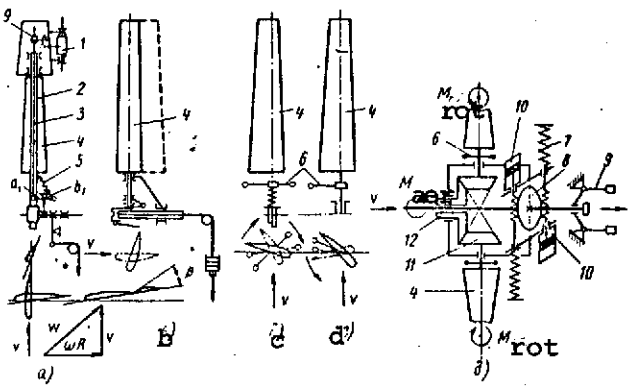


Fig. 108. Basic system for regulation of high-speed wind machines: a. stabilizer; b. safety sail; c, d. centrifugal-aerodynamic; e. moment-centrifugal; 1. stabilizer; 2. spar; 3. regulator rod; 4. blade; 5, 7. springs; 6. compensating weights; 8. toother wheel; 9. centrifugal regulator; 10. damper; 11. blade gear; 12. kinematic connection gear.

If an increase in the windmill's rotation frequency is higher than calculated, rod 3 of the regulator (Fig. 108, a) moves and turns the stabilizer through a system of levers (Fig. 109, position IV). Due to the destruction of the equilibrium between moments M_{bl} and M_{st}, the blade rotates in a direction decreasing angles α, and the lifting force is decreased, leading to a reduction in the windmill's rotation frequency. If the stabilizer is reset in its initial position I, the moment on the blade becomes negative and the windmill stops. /18

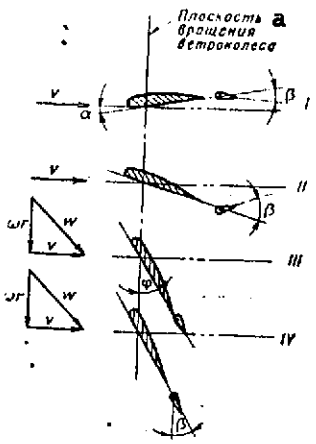


Fig. 109. Operation of stabilizer regulation: I. motor stopped; II. starting position; III. work before beginning of adjustment; IV. work during the process of adjustment.

Key: a. Plane of windmill rotation.

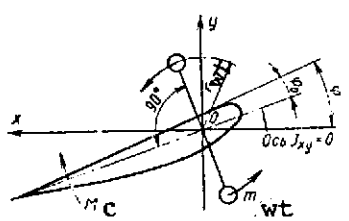


Fig. 110. Determining the centrifugal moment of inertia of a blade.

The centrifugal forces of the masses of the blade and stabilizer, whose moment M_c hinders rotation of a blade under the action of aerodynamic forces also participate in regulation:

$$M_c \approx J_b \omega_{2m}^2 \frac{\sin 2\phi_b}{2}, \qquad (149)$$

where J_b is the moment of inertia of the blade with its stabilizer relative to the axis of the spar; ϕ_b is the angle between the chord of the blade and the plane of rotation of the windmill.

Weights, whose moment of centrifugal forces is equal in size and opposite in sign to the moment M_c (Fig. 110) are installed for total compensation of the centrifugal moment of inertia on the blade. With partial compensation (undercompensation) of the summary moment of centrifugal forces of the masses of the blade and weights strive to set the blade in the plane of rotation of the windmill, and with overcompensation, conversely, it strives to set the blade perpendicular to the plane of rotation [8, 30].

The system of regulation examined is included in the indirect action type of regulation, where the stabilizer plays the role of an aerodynamic servomotor. It provides high regulator sensitivity and proper evenness of motor rotation, which /182 also allows it to be used for wind electric stations. The D-12, D-18 and D-30 motors have stabilizer regulation.

The safety sail regulation system (a system of A. G. Ufimtsev and V. P. Vetchinkin), in which the blade rotates due to the combined action of the aerodynamic moments arising on the blade (main forces) and the centrifugal forces of the mass of the blade itself. This regulation is one of direct action, since the work necessary for resetting the working organ is performed by the sensing element itself.

The regulation system acts in the following manner (see Fig. 108). The blade is freely fitted on the spar so that its

axis passes near the leading edge of the blade and equally acting aerodynamic forces create a moment which attempts to turn the blade in the wind vane position. With a wind velocity $v < v_c$, the blade, due to a support, rests at an angle of $\phi = \phi_{opt}$ and is held in this position with a weight or with springs. With an increase in wind velocity, moment M_{aer} relative to the axis of the spar increases, and the blade turns so that overloading of the motor and an extreme increase in windmill rotation frequency are not attained. For stopping the windmill, the blade is forcibly turned into the wind vane position using the cable of the starting and stopping system.

This system of regulation has great flexibility in tuning, allows a relationship between the values of M_{mo} and ω_{wm} to be realized, and possesses the aerodynamic damping which is enhanced by the stability of direct regulation even without considering friction. Changing the relationship of the aerodynamic and centrifugal forces, the size of the weights and the kinematics of the regulator, it is possible to achieve the required regulation schematic. This system is used in the D-10 and D-12 wind electric stations, which are built in the Kursk region, and in the 1D-18 of TsAGI. /183

The stabilizer and safety sail systems of regulation are used for wind electric stations which work in isolation or parallel with a non-wind electric station. However, in the latter case it is necessary to use additional devices in the windmill's layout to protect the motor from high overloads during an increase in wind velocity, which complicates the machine and reduces its coefficient of efficiency.

The centrifugal-aerodynamic regulation system, with which the blade rotates relative to its own longitudinal axis against the flow, i.e. in the direction toward decreasing the angles of inclination (the system of Ye. M. Fateyev and G. A. Pechkovskiy) and with the direction of the flow, i.e. in a wind vane position, due to the action of the centrifugal forces of regulator weights and the masses of the blades (see Fig. 108, c and d).

Before the beginning of regulation, the sum of moments $M_{cf} + M_{aer}$ is less than the moments of the forces of spring flexion. The position of the blades is fixed at a set angle ϕ_b and they are as if rigidly fastened to the hub of the windmill. With an increase in the windmill's rotation frequency due to a decrease in the load or an increase in wind velocity, the blades, under the action of aerodynamic and centrifugal moments of inertia which overcome the force of the spring, rotate by some angle corresponding to equilibrium of the moments. This will lead to a decrease in moment M_{mo} and to reestablishment of the set rotation frequency. For stopping the windmill, the blades are rotated to an angle at which moment M_{mo} becomes equal to zero.

During regulation by turning the blade along the direction of the flow for total stoppage of the windmill, it is necessary to turn the blade by 80-90°, and during regulation for turning it against the flow, it is necessary to turn it by only 20-25°. However, frontal pressure is increased with this.

Regulation by turning the blade due to the action of a centrifugal regulator is also direct action regulation. It is usually used for motors having a power of up to 6 kW, when it is necessary to have a relatively low unevenness in the windmill's rotation frequency with changes in wind velocity and load. For large diameter windmills, the best characteristics can be obtained by turning the blade using the indirect regulation layout, in which a hydraulic or pneumatic system with corresponding boosters are used in addition to the centrifugal regulator. This also creates the best dynamic stability for the regulator.

While for low-power machines the requirements for a windmill regulator come down primarily to limitation of the rotation frequency ceiling, the medium-power motor working with a generator /184
and pump must have a regulation system whose functioning is more complex. In particular, when utilizing such machines for irrigation, drainage or reclamation, groups of machines working in parallel with each other or with non-wind machines are effectively used. The task of limiting the moment developed by the wind motor becomes one of the most important in this case.

Especially rigid requirements are laid down for a regulator with parallel operation of wind machines and a network. The most important thing in this case is calibration with a measuring organ of the direct deviation of that value which is the original cause of destruction of the working rate, i.e. wind velocity or rotational moment [4]. With this it is necessary to provide sufficient power to the measuring part to get by without an intermediate booster. One of the technical solutions of this problem is the moment system of regulation proposed by the author. The windmill works in a zone of high flow turbulence, characterized by low concentration of energy. Therefore, the wind regulator[1] (device acting in dependence of a change in the pressure or velocity of the flow) is difficult to use: it cannot consider the combination of all factors determining the operating rate of the machine, and it will be clumsy and not very effective. Calibration of moment M_{mo} and utilization of it as a parameter controlling the regulator satisfies the basic requirement: regulation is distinguished by the high power of the measured portion, and calibrates and reflects in complex all the

[1] A wind regulator can be made in the form of a plane or small airfoil located in front of the windmill which absorbs the pressure of the flow and acts on the mechanism controlling the angle of rotation of the blades.

factors determining the operation of the machine (v, ω_{wm}, M_r) and going into the differential equation of its motion

$$J_\Sigma \frac{d\omega}{dt} = M_{mo} - M_r - M_{in.r}, \qquad (150)$$

where $M_{in.r}$ is the moment of internal resistance of the drive (transmission).

These considerations determine the principles for constructing the system of combination moment-centrifugal regulation, realized with the rotational moment of the aerodynamic forces of the windmill and the centrifugal forces of the regulator. The sensitive element of the moment part of the regulator reacts directly to a change in the rotational moment and transmits a command to the calibrating organ -- the turning blades. The centrifugal part reacts to a change in windmill rotation frequency and also transmits a command to the blades. In this way, the basic setting and adjusting parameter is first of all moment M_{mo} and therefore the danger of overloads is decreased and the machine can be used in any layout: isolated, together with a non-wind installation and parallel operation with an electric power station or network. /185

The principle of regulation consists of the following. The moment regulator in the hub of the windmill, which is freely fastened on the main shaft, reacts directly to a change in moment M_{mo}. The rotating blades are simultaneously absorbing and executing elements of the system. The effect of windmill rotation frequency on the angle of blade setting comes down to a minimum of total compensation of the centrifugal moment on the blade and the use of a blade with a symmetrical profile, in which values for rotating moments M_{Oz} of aerodynamic forces relative to the longitudinal axis Oz are low. It can be practically considered that the regulator reacts only to a change in moment on the windmill shaft. However, with the load disengaged and with $v > v_{n.reg}$, when $M_{mo} >> M_r << M_w$, the moment regulator cannot exhibit its properties. Therefore, there is also a centrifugal regulator in the system, which, changing the angle of blade settings, protects the windmill from overwearing.

The design layout is executed so that the windmill hub 4 (see Fig. 42) can rotate relative to shaft 5 with toothed wheel 13 on roller bearings. The toothed wheel is engaged with two toothed racks 14, which are in turn connected with the hub through flexible elements (springs) 15. The blades are fastened in the hub on bearings. On the end of the spar is gear 16, which is engaged with kinematic connection gear 3.

While the inequality $M_{mo} < M_w$ is maintained, the windmill and hub rotate as a single unit with the shaft. Their motion is

determined by the aerodynamic characteristics of the windmill at $\phi = \phi_w$, so that the blades are set on a support. The preliminary tension of the flexible elements of the moment regulator correspond to M_w. With an increase in moment M_{wm} (for instance, in parallel operation of the machine with a circuit, if $v > v_w$, with an increase in moment M_r and at emergency rates) the hub turns relative to the shaft and sets the blades at an angle at which moment M_{mo} is decreased. If the windmill rotation frequency increases, the centrifugal regulator comes into play, rotating the blades through the kinematic connection gear and reestablishing the equilibrium position.

For improvement of starting characteristics of the machine, the blades are automatically rotated to the optimum angles during starting and acceleration. With proper parameters for the regulator, deviations of moment M_{mo} from calculated values do not exceed 8-10%, and those of windmill rotation frequency are no greater than 20%. A low unevenness of pace creates the necessary conditions for operation of an electric drive.

The use of rotating blades complicates design and therefore designers strive to stay away from regulators entirely or essentially simplify them. One of the solutions of this problem is the creation of windmill with rigidly fastened blades which possess the property for aerodynamic self-adjustment [58]. The use of these windmills improves the design of the machine and increases its operational reliability. These systems are most effective at low initial moments of load resistance or when relieving mechanisms are present (this is caused by the necessity for rotating the blade to high starting angles), and also with a relatively high permissible rotation frequency unevenness /186

$\delta_\Sigma = \frac{\omega_{max} - \omega_{min}}{\omega_{av}}$. High-speed machines ($Z_n \geq 6$) with rigidly fastened blades can also be used for parallel operation with a network with direct current frequency, with which high overloads do not arise.

The effect of self-adjustment, i.e. the capability of limiting the moment with $v > v_w$, is achieved with any layout utilizing a wind energy-producing machine combining the properties of a normal windmill and a radial fan into a single system (Fig. 111). This windmill consists of a hub and blades with longitudinal ventilation channels connected with the intake hole in the hub. The channels have valves with a spring or weights (diagram in Fig. 111, a). In accordance with the diagram in Fig. 111, b, the hub has a panel (cone), which is pressed against the edges of the intake hole by the spring. The tension of the spring is adjusted so that with $v \leq v_w$ the holes are totally closed and the windmill works in accordance with normal aerodynamic characteristics.

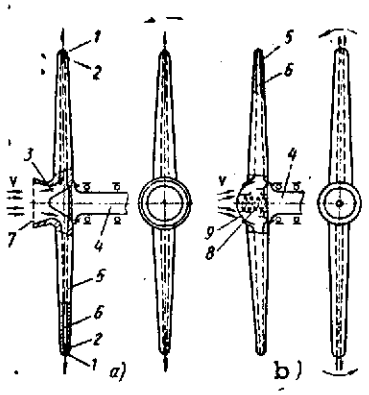


Fig. 111. Diagram of windmill possessing the property of self-adjustment: 1. valve; 2, 8. springs; 3. intake hole; 4. shaft; 5. blade; 6. channel; 7. screen; 9. panel.

In the diagram in Fig. 111, a, where the main set parameter is ω_{wm}, and increase in velocity v or, conversely, a reduction in moment M_r will lead to an increase in $\omega_{wm} > \omega_{wm.w}$. The valves, overcoming the force of the springs, move under the action of centrifugal forces, the passage sections between the seat and valve open and air begins to be thrown out in the radial directions. From this moment, the windmill also work simultaneously as a fan, flinging air along the channels. The coriolis forces of inertia of the air masses in the channels are overcome by the moment for aerodynamic forces arising on the blades. /187

With a moment M_r = const, a further increase in velocity v causes some increase in the values of ω_{wm} and P_c of the masses of the valve. The valves move and the passage sections increase. An increase in passage sections, and also in velocities of the flow in front of the hub increase the fan effect, which leads to stabilization of values N_{wm} and ω_{wm}.

In the diagram in Fig. 111, b, the given parameter is the velocity of the flow in front of the windmill. With an increase in velocity v, frontal pressure on the cone increases and the slot between it and the walls of the hole grows larger. Due to the suction created by centrifugal forces, the air is flung along the channels and the braking effect of the fan arises. The power expended by the fan depends primarily on the section of the valves and on frequency of rotation. Due to the pressure created by the flow, the characteristics of $M_{fan}(v,\omega_{wm})$ can shift relative to curves $M_{mo}(v,\omega_{wm})$. In the diagram on Fig. 111, b, the unevenness of rotation is greater than in the diagram on Fig. 111, a.

The power N_{wm} developed by the windmill, and the second's work A_{fan}, expended on shoving the air through the valves, are equal to

$$N_{wm} = \pi R^2 \frac{\rho v^3}{2} \xi$$

$$A_{fan} = Q\rho cu = F_{v\Sigma}\omega\rho cu = F_{v\Sigma}(u - c)\,\rho cu, \qquad (151)$$

where $F_{v\Sigma}$ is the total area of the valves; $w = u - c$ is the relative velocity of air exhaust from the valves; c is the velocity of the air in the channels; u is the circular velocity at the median radius of the blade.

Varying the value of $F_{v\Sigma}$ and other parameters, it is possible with $v > v_{n.reg}$ to reach the equation

$$M_{mo} = M_r + M_{fan}, \qquad (152)$$

which provides self-regulation of the windmill.

The designs of regulators for some motors are shown in Fig. 112 and in Chapter 6. When planning regulators, one should strive toward reduction in the forces of friction, the possibility for compensating the weight unevenness of the windmill, eliminating large clearances, sufficient rigidity in the elements of kinematic connection of the blades and others.

Dynamic processes, which frequently have a decisive effect on the interrelationship of parts and mechanisms and the stability of their operation, play a large role in high-speed machines. For a two-bladed motor with centrifugal-aerodynamic regulation, intensive transfer rates arise during operation of the regulator and in the configuration of tangential blow-by of the windmill during reorientation of the head.

The operation of the regulator for the "Vikhr'" machine is captured by a photo sequence (Fig. 113), with $\bar{v}$ = 7.6 m/sec and $\omega_{wm.mom}$ = 38.1 rad/sec. Frame a shows the misalignment of the carriage, which is fastened onto the pole of the starting and stopping system. If misalignments are constant, this may be a reason for breakage of the pole. The carriage is distorted when the blades are in the vertical position; there is no misalignment in the horizontal position. /189

The following diagram of this phenomenon can be proposed: with high angular gradients of the wind velocity vector and a low velocity of head reorientation, oblique blow-by of the windmill occurs over the course of some period of time, and therefore the upper blade, moving "away from us" (behind the plane of the frame), and the blade rotating "toward us" are located in essentially different aerodynamic conditions. Triangles of velocities in similar sections and rotating moment M_{zz} of the blades will also be different. In the case examined, moment M_{zz} is greater on the upper blade. This indicates that the stream has turned counterclockwise (looking from above). Misalignment of the carriage causes a difference $\Delta\phi$ in momentary values of the angles of blade setting, which creates an additional unevenness and vibration.

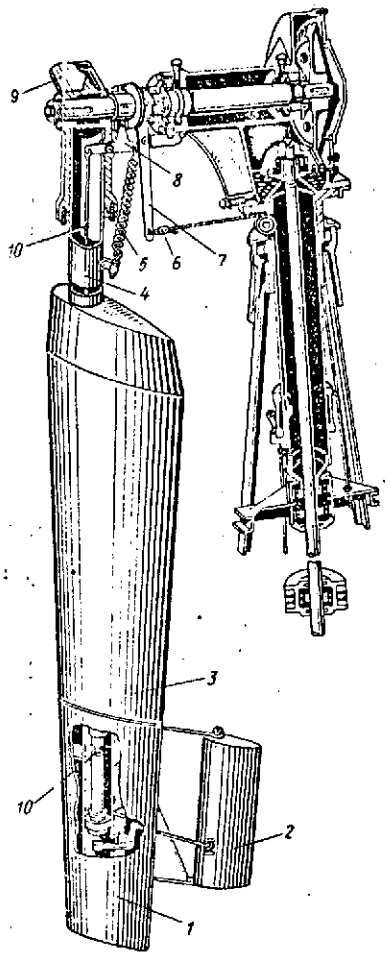


Fig. 112. Construction of regulator for the D-12 high-speed machine: 1. rotating blade; 2. stabilizer; 3. blade; 4. spar; 5. spring; 6. cable; 7. stopping lever; 8. double armed lever; 9. hub; 10. rod.

Rigid fastening of the carriage and the pole eliminates these phenomena.

The calculation of static and dynamic characteristics of regulation usually represent the maximum difficulty and have a number of singularities for each type of regulator. We will deal only with the common problems of the statics and dynamics of the regulation, referring readers who are interested in this in more detail to special literature [8, 30, 37, 61].

The term static characteristic of regulation is usually understood to mean the dependency of the moment or rotation frequency of the windmill on any parameter (angle of blade setting, wind velocity, movement of regulator weights, and others), with which the calculated or experimental aerodynamic characteristics are used. For examining the statics, equations of the system's equilibrium are composed. Thus, for a motor with an aerodynamic system of regulation by rotating the blade, with v = const, the following two equation exist

$$M_{mo}(Z,\phi) = M_r;$$

$$M_{aer}(Z,\phi) - M_{cf}(\omega_{wm},\phi) = M_{wt}(\phi), \quad (153)$$

where M_{wt} is the moment from the regulator weights.

Setting the value of M_r and solving this system, the dependency between the values is established and the static properties of the motor are developed. For this, the possible positions of equilibrium of the blade and regulator are located while the wind motor is working at various rates, i.e. at various wind velocities and loading moments. Since the equation is not solved analytically, it is necessary to construct curves $F(\phi) = (M_{aer} - M_c)$ for various values of v; points of their intersection with the curve $M_{wt}(\phi)$ will also be the solution sought (Fig. 114). The points of intersection of the loading parabola with the aerodynamic characteristics of the windmill, obtained for various angles of ϕ, will /190

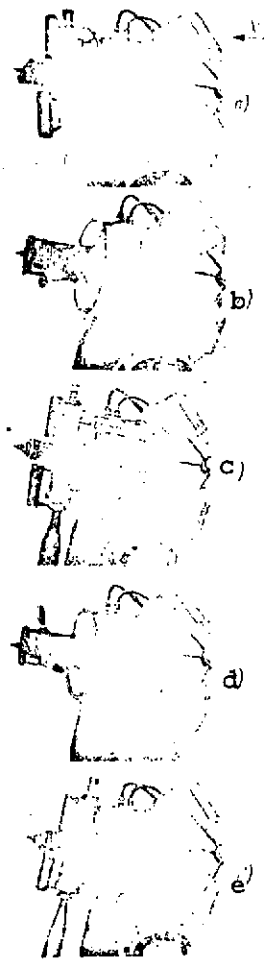


Fig. 113. High-speed photos of operation of a regulator for a high-speed two-bladed machine with a windmill diameter of D_{wm} = 4 m.

yield the corresponding working rates characterized by the value Z (Fig. 115) in exactly the same way.

For a moment regulator with three blades at v = const and $\omega_{wm} = \omega_{wm.nom}$ = const, there are three equations

$$\left.\begin{aligned} M_{mo}(\varphi) &= M_r \\ M_{mo}(\varphi) &= M_{sp}(\alpha_1) + M_b(\varphi) = \\ &= M_{sp}(0) + k_1\alpha_1 + 3k_f[M_{c.f}(\varphi) + M_{b.aer}(\varphi)]; \\ \varphi &= \varphi_z + k_f\alpha_1. \end{aligned}\right\} \qquad (154)$$

where $M_{sp}(\alpha_1)$ is the moment created by the regulator spring relative to the axis of rotation of the windmill; α_1 is the angle of rotation of the hub relative to the main shaft; M_b is the moment created by the blade relative to the main shaft; k_f is the transmission function of the regulator; k_1 is the rigidity of the regulator spring.

The moments $M_{mo}(\phi)$ and $M_{b.aer}$ are determined according to the characteristics of the windmill and blade or according to formulae (136) and (156). It is convenient to represent the characteristics not in absolutes, but in relative values, for instance, moment $M_{mo}(\phi) = \frac{M_{mo}(\phi)}{M_n}$, i.e. in proportions of the nominal value of moment M_n and the wind velocity in proportions of the calculated (nominal velocity).

During rotation of the blade, the aerodynamic forces are changed (Table 24). /191

TABLE 24. ORIENTING VALUES OF COEFFICIENT C_y AND C_x OF A BLADE

Coefficient	Angle of attack α in degrees												
	−10	0	10	20	30	40	50	60	70	80	90	100	110
C_y	−0,1	0,7	1,3	1,25	1,05	0,7	0,62	0,5	0,4	0,2	0	−0,2	−0,5
C_x	0,02	0,02	0,12	0,28	0,42	0,65	0,82	1,0	1,1	1,12	1,11	1,05	0,9

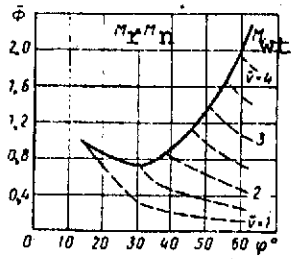


Fig. 114. Characteristics of $F(\ ) = M_{aer} - M_c$ (according to the data of K. P. Vashkevich); $\bar{F} = \frac{F(\phi)}{F(\phi_0)}$; $\bar{v} = \frac{v}{v_c}$

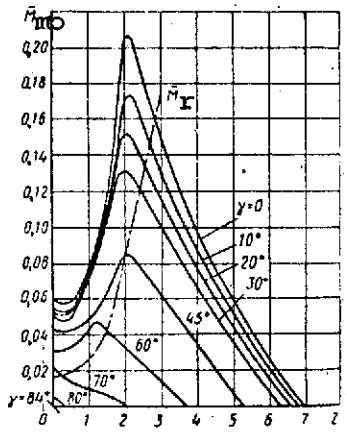


Fig. 115. Determining working rates of a regulator:

$\bar{M}_r = \frac{M_r}{M_{r.n}}$

During rotation of a blade by a stabilizer having an area f_{st}, the value of the rotational moment of the latter will be

$$M_{st} = C_R f_{st} \frac{\rho v^2}{2} l \pm \frac{G_{r.b}}{9.81} \omega_{wm}^2 r_{av} \mu r_{br} \pm M_{b.aer} \quad (155)$$

where C_R is the coefficient of resultant aerodynamic forces for low angles of attack ($C_R = \sqrt{C_y^2 + C_x^2}$ can be used); l is the distance between the axes of the stabilizer and turning portion of the blades; $G_{r.b}$ is the weight of the rotating portion of the blades; μ is the coefficient of friction in the bearings; r_{br} is the average radius of the bearings; r_{av} is the distance from the center of gravity of the rotating portion of the blade to the axis of the windmill.

During the regulation process, the stabilizer is rotated by the action of the centrifugal forces of the weight and a rod

$$P_{ef} = \frac{\omega_{wm}^2}{g} (G_1 r_1 + G_2 r_2 + \ldots + G_n r_n),$$

where $G_1, \ldots, G_n$ is the weight of corresponding elements of the regulator; $r_1, \ldots, R_n$ is the distance of the elements from the axis of rotation.

The action of aerodynamic forces arising on the stabilizer can be ignored due to their smallness.

The action of centrifugal force is equalized by the tension force of the spring p_{sp}. Consequently

$$P_{cg} a_1 = P_{sp} b_1,$$

where a_1 and b_1 are the arms of a lever, to the ends of which are fastened the rod and regulation spring (Fig. 108, a).

For rotating the blade under the action of aerodynamic forces arising on its surface, the axis of the spar must pass close to the leading edge of the blade. The moment rotating the blade relative to the spar axis is equal to /192

$$M_{b.aer} = C_y S_b \frac{\rho}{2} (\omega_{wm}^2 r_m^2 + v^2) x_1, \quad (156)$$

where $r_m = \sqrt{\frac{R^2 + Rr_0 + r_0^2}{3}}$ is the conditional radius of the airfoil; $x_1 = x - a_1$ is the distance from the axis of rotation to the point of application of the equalizing forces; a_1 is the distance from the leading edge of the wing to the axis of the spar.

It is possible to take $x = 0.33b$ with $\alpha = 30°$; $x = 0.57b$ with $\alpha = 40°$; $x = 0.77b$ with $\alpha = 50°$; and $x \approx b$ with $\alpha = 60°$.

During regulation by rotating the blade due to the action of the inertial forces of its mass and additional weights, the rotating moment M_{rot} is determined according to formula (149). The maximum value of moment M_{rot} corresponds to an angle $\phi = 45°$

$$M_{rot\ max} = \frac{J_\Sigma \omega_{wm}^2}{2}. \quad (157)$$

Static stability of the regulator is provided if, during deflection of the blade at a small angle, all the moments going into equations (153), (154), and (159) attempt to turn it into its previous condition of equilibrium. FOr this, it is necessary that

$$\frac{\partial \overset{*}{M}_{mo}}{\partial \phi} - \frac{\partial \overset{*}{M}_b}{\partial \phi} - \frac{\partial \overset{*}{M}_{sp}}{\partial \phi} - \frac{\partial \overset{*}{M}_r}{\partial \phi} < 0. \quad (158)$$

For providing normal operation, the regulator must be not only statically, but also dynamically stable.

With v = const, the power of the motor depends on the rotation frequency of the windmill and the angle of blade setting. During regulation by turning the blade by, for instance, the effect of aerodynamic forces, values of ω_{wm} and ϕ are independent and they should be regarded as generalized coordinates characterizing the working rate of the motr. The motor, together with

the regulator, is a mechanical system with two degrees of freedom, and therefore two equations are sufficient for its investigation

$$J_{wm} = \frac{d\omega_{wm}}{dt} = M_{mo} - M_r;$$
$$J_{zz}\frac{d^2\phi}{dt} + k_1\frac{d\phi}{dt} = M_{b.aer} - M_{cf} - M_{reg}. \quad (159)$$

where J_{wm} and J_{zz} are moments of inertia of all bodies kinematically connected with rotation of the windmill and blades, adjusted to their axes; k_1 is the derivative damping moment of aerodynamic forces.

Two equations for regulation by rotating the blades, controlled by a centrifugal regulator, can be written similarly. Motors having a stabilizer or moment regulation are systems with three degrees of freedom; in the first method of regulation, an additional equation for small oscillations of the regulator or, likewise, motion of the stabilizer must be written; in the second, an equation for rotation of the windmill hub relative to its shaft must be written. /193

With consideration of kinematics and transmission ratios of the regulator mechanism links, expressions for adjusted values for inertial moments of the weights and other parts are written, and using extracted aerodynamic moments, moments M_{mo}, $M_{b.aer}$, M_{cf} and M_{reg} are computed. Besides external loads, which must be functions of the values ω and t, constant values (dry friction and others) also go into the moment of resistance.

The derivative damping moment of aerodynamic forces is approximately equal to [8]

$$k_1 = \int_{r_0}^{R} k_0 \frac{\rho\omega}{2} b^3\, dr = \frac{\rho\omega}{8}\int_{r_0}^{R}\left[(1-2\bar{x}_0)^2\frac{dC_y}{da} + \frac{1}{4}\left(2\pi - \frac{dC_y}{da}\right)\right]\times$$
$$\times (1-e)\, b^3\sqrt{1+Z_u^2}\, dr, \quad (160)$$

where $\frac{dC_y}{da}$ is the derivative for an airfoil with length $\lambda = 2\frac{R - r_0}{b_1 + b_0}$; $\bar{x}_0 = \frac{x_0}{b}$ is the relative distance between the axis of rotation of the blade and its leading edge.

A number of wind machines are characterized by low transmission rigidity and high moments of inertia in their driven parts, therefore torsional oscillations can arise and motion in the system should be examined with consideration for twisting of the transmission. In this case, the first equation in system (159) takes on the appearance

$$J_w \frac{d\omega}{dt} - M_{mo} = -q_s \theta_d$$

and a third equation appears

$$J_m \frac{d^2\theta_d}{dt^2} - q_s \theta_d = -M_r,$$

where q_s is the adjusted torsional rigidity of the drive; J_m is the moment of inertia of the driven portion, adjusted to the windmill shaft; θ_d is the angle of torsion of the shaft (drive).

According to data of K. P. Vashkevich, the final form of the system of equations for the case of aerodynamic regulation is the following /194

$$\left.\begin{aligned} J_w \frac{d\omega}{dt} - M_{mo} &= -q_s \theta_d \\ J_m \frac{d\omega}{dt} - J_m \frac{d^2\theta_d}{dt^2} - q_s \theta_d &= -M_r; \\ J_{zz} \frac{d^2\varphi}{dt^2} + k_1 \frac{d\varphi}{dt} &= M_{b.aer} - M_{c.f} - M_{reg} \end{aligned}\right\} \quad (161)$$

Solving these equations, we locate the functions ω, ϕ and θ_d.

Slight deviations in the moving system from the established rate are usually examined, and therefore portions of characteristics reflecting the work of the regulation mechanism can be replaced with sections of tangents. An unexpected wind gust, breakage or engagement of the load can be a source of disturbance. Having written equations establishing the rate, it is possible to determine whether the system is static or astatic. In the first case, constant values of $\Delta\omega$ and $\Delta\phi$ correspond to each constant value of a disturbing force.

Using the special criteria of the regulation theory and composing the characteristic equations of this sytem, the problem of dynamic stability of regulation is solved. In particular, for this it is necessary that all coefficients of the equation are positive (the so-called Gurvitz criterion).

Besides stability, the regulation system must possess definite qualitative indices (accuracy of regulation, action speed and others). Therefore, it is necessary to examine the character of the transfer process during the time of regulation and establish its most advantageous duration. It is important to provide conditions in which oscillations of the regulator are rapidly damped so that the phenomenon of resonance from coincidence of the frequencies of oscillations of the system itself and those of disturbing forces does not occur.

Fig. 116 shows the work of a centrifugal-aerodynamic regulator of the "Vikhr'" machine. Dynamic stability was determined in a wind tunnel by the method of running the windmill with established flow velocities of v = 8-25 m/sec and moment $M_r = 0$, /195 which, from the point of view of stability, corresponds to the worst rate. From Fig. 116 it is evident that only at v = 8 m/sec does the regulator kick in with some overregulation. The regulated system is dynamically stable at all rates. The first travel of the sleeve corresponds to compression of the starting spring and rotation of the blades with $\phi_r = 34°$ up to $\phi_{work} = 8°$. With the second movement, the main spring is compressed, and the blades are rotated up to $\phi = -4°$, corresponding to idle at $v = v_c$. The small self-oscillations, with an amplitude $\Delta\phi = \pm 0.5°$ and a period $T \approx 3$ sec observed in some cases did not deteriorate the operation of the regulation system and did not represent any danger for the strength of the machines.

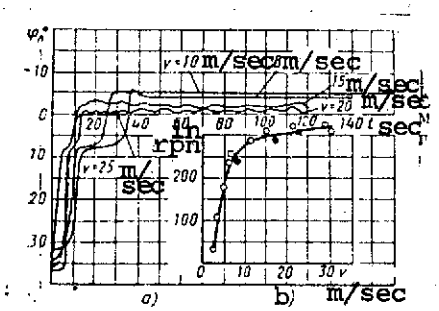


Fig. 116. Operation of the centrifugal-aerodynamic regulation of the "Vikhr'" machine: a. transfer processes; b. static characteristic.

In conclusion we note that in wind machines, two types of remaining unevenness of pace $\delta = \frac{\Delta\omega_{wm}}{\omega_{wm.av}}$ should be differentiated: 1) according to load and 2) according to wind velocity. The first evaluates the increase in ω_{wm} (with respect to $\omega_{wm.av}$) with removal of the nominal load, and the second yields the value of $\omega_{wm\ max}$ with a definite multiplicity factor in the increase in wind velocity (with respect to v_c).

Chapter 10. Calculation and Design of Wind Motors

The majority of components and mechanisms of a wind machine, primarily a wind motor, works at changing rates, caused by changes in the load, the energy of the airflow and the vector of its velocity, and the action of the regulator. The most responsible components,

whose loading determines the reliability of the machine, are the windmill, supporting device of the head, the orientation mechanism, transmission and tower. The calculation of transmission mechanisms (reduction gears, shafts) is accomplished according to the usual methodologies employed in machine construction, however with consideration of overloads, which can arise in connection with the peculiarities of a motor regulation system and the characteristics of the working machine.

Considering the difficult conditions of operation, primarily the effect of the pulsing flow, including those with storm velocities, and the absence, as a rule, of constant observation on the part of servicing personnel, special attention should be paid to the reliability of the machine as the basic factor, largely determining the effectiveness of its use. By comparison with these questions, the problem of further reducing metal consumption, although important, nevertheless has a subordinated value which, as a rule, eases creation of a reliable machine.

We will evaluate the reliability of a machine with the index

$$k_r = \frac{T_0}{T_0 + T_{\Sigma id} - \Sigma_{id}^{t'}}, \qquad (162)$$

where T_0 is the overall duration of work; $T_{\Sigma id}$ is the summary idle time; $\Sigma_{id}^{t'}$ is the idle time due to windless conditions, forced stopping of the machine, the absence of the requirement for power or full charging of a storage battery device. /196

The size of k_r depends on the layout and design of the machine, the quality of materials and the technology of manufacture, operating rates and conditions and, subsequently, is a function of regional conditions, primarily wind conditions. The storage of energy and reserving of power should be seen as one of the factors for increasing the reliability of all wind installations and, in the final analysis, for better provision of energy (water) or a product of reworking to the consumer.

Many factors determining reliability are random, and it is therefore expedient to evaluate them statistically. At the same time, the optimum level of reliability can be determined according to summary expenditures, locating the minimum of the expression

$$\bar{E}_\Sigma = \bar{E}_0 + \bar{L} \to \min, \qquad (163)$$

where $\bar{E}_0$ is the expenditures on acquiring the machine, its serving and repair; $\bar{L}$ is the losses of the consumer from interruptions in the power supply (Fig. 117).

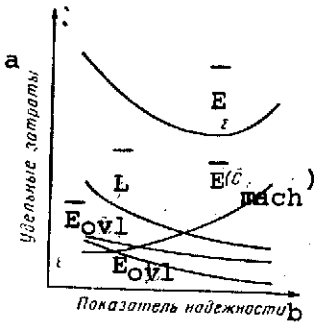


Fig. 117. Determining optimal reliability of a machine.

Key: a. Specific expenditures; b. Index of reliability

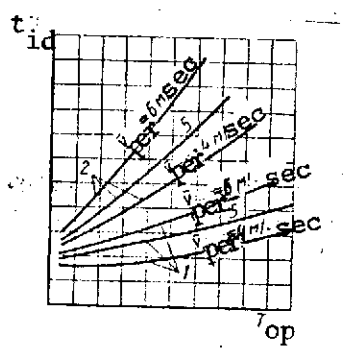


Fig. 118. Effect of external factors on longevity of idle periods of a machine for technical reasons: 1) H_Σ = 10 m; 2) H_Σ = 20 m.

For machines of a single type, the time of stoppages for technical reasons, which goes into the summary time $T_{\Sigma id}$, depends both on objective factors (quality of manufacture, deficiencies in construction, the action of hurricane winds), and on nonobjective ones (quality of assembly, system of servicing). /197 At the same time, the probability for an increase in idle times with all other conditions being equal increases as the period of the machine's operation grows longer, the depth of the water source or power of the load connected grow larger, or the wind rate of the region grows more intensive (Fig. 118). The latter has an essential effect on the reliability of the machine.

§ 1. Loads on Elements of a Wind Motor

Calculations for the strength of elements of a wind motor consist of determining constant and variable stresses in the characteristic sections of these elements under varying conditions of loading, i.e. at normal operational and some emergency rates. The calculation of flexible oscillations, primarily those of the blades, as well as those of stability of the tower at rates close to resonance rates, must be an inseparable part of calculations for dynamic strength. The solution of these problems requires knowledge of actual loads acting on the motor and dynamic processes occurring during its interaction with the flow, especially during regulation rates.

Although practical methods for computing some elements of wind machines have been developed, they do not always allow the necessary accuracy to be obtained.

Investigations have shown that a change in loads on the elements of a machine is a random stationary process: with a sufficiently long record of stresses, the static properties of a curve describing pulsation remains practically unchanged. Therefore, a single realization of a random function, written over the course of a long time interval with any of the ranges of wind velocities and gradations of loading, is sufficient for computing all the necessary characteristics of the process. A change in the velocity of the airflow is a random process, which can be divided into an infinite

harmonic series, with which the amplitude of each of its harmonics will be a random value. For short time intervals (10-15 min), the record of $v(t)$ can be considered as possessing an ergodic property.

The major loads on the windmill, tower, and lateral plane are the loads brought about by the forces with which the airflow acts on these elements. The nominal values of forces arising in the established flow are determined by the aerodynamic calculation. Under natural conditions with rapid changes in the airflow, these forces can briefly increase, which is explained not only by the structure of the airflow during gusts, but also by the inertia-possessing property of the regulation system. Determination of overloads arising during this, i.e. the relationship of maximum forces and forces acting during nominal rates, can be based on calculations and on experimental data.

Aerodynamic loads are connected with the optimum values for /198
for maximum wind velocities v_{max}, which are determined during calculations of parameters of the machine (Chapter 4). Usually v_{max} = 30-40 m/sec. At the same time, for a machine with a power of more than 10 kW, according to GOST [All-Union State Standard] 2656-55, it is recommended that the v_{max} velocity be taken as equal to 40, 50 and 60 m/sec respectively for regions with a velocity $\bar{v}_a$ up to 5, 5-7 and over 7 m/sec. The coefficients of overloads k_{ovl} with a duration of a minute are set at double and not differentiated according to the types of machines. Under actual conditions, the coefficient k_{ovl} is a function of the type of drive, regulator, the moments of inertia of the driving and driven parts of the motor, etc.

Among the forces determining the basic loads on elements of a motor, the main ones are: forces arising from the rotational moment of the windmill; forces of gyroscopic origin, centrifugal forces and, finally, the weight of the windmill, head, tail and lateral plane. These forces are determined according to formulae, which are presented in sections dealing with calculation of the corresponding components.

Along with the main loads, additional and special (incidental) actions of the flow on the motor should be differentiated. This is especially important to keep in mind if calculation of the design is conducted not on the method of allowable stresses, based on introduction of coefficients of reserves and, strictly speaking, hindering the maximum total utilization of the material, but with the method of limiting conditions, guaranteeing that over the operating time with the selected maximum wind velocities and the most difficult combination of loads, not one of the limiting conditions will take place, i.e. breakage will not occur.

In accordance with construction norms and rules and existing classifications, we will include wind machines in constructions of

class IV, and wind loads will be seen not as brief in time, but as a normal operation influence of changing intensity and definite dynamics, evaluated with the coefficients of gustiness. With this, the average intensity (power) of gusts must be expressed with equation (5), and the probability for high velocities must be expressed with formula (4).

A wind machine, up to 40% of the time, develops power which is less than the calculated power, and does not work at all for 20-25% of the hours in a year. Therefore, when calculating for endurance of bearings for components, toothed transmissions, blades and shafts, it is not economic to consider the load constant, equal to P_1, but coefficients of changeability in the working rates k_{rt} should be introduced [57].

Considering this and using the hypothesis of linear summarization of manageability, which expresses that the failure of a design occurs when the summary damageability is

$$\Delta\bar{N}_{\Sigma}=\sum\frac{\Delta N_i}{N_i}=1, \qquad (164)$$

where ΔN_i is the actual (calculated) number of loading cycles with amplitude σ_i, corresponding to the determined working rate of the machine; N_i is the number of cycles before failure. With uninterrupted maintenance of the level of loading with the same amplitude, it is possible to establish the resource (longevity) of the most highly loaded element. The value ΔN_i is called the damageability of the design at a working rate with amplitude of loading σ_i. /199

§ 2. Selection of the Type of Windmill

The major requirements outlined for a windmill consist of providing the initiation of the work of the machine at set minimum wind velocities, for which the aerodynamic moment M_0 during run-up must be higher than the moments of resistance created by the drive mechanism and working machine. The coefficient of utilization of the wind's energy must be relatively high (0.35 and higher), and the system for regulating rotation frequency and power must provide reliable protection of the windmill for overloads and overspeeding, as well as assuring operation of the machine with a certain evenness in rotation frequency. Thus, when working with a piston pump, the maximum rotation frequency must not exceed the calculated one by more than 40%. In pneumatic machines, unevenness in rotation frequency does not have a great meaning, and it is important only to protect the windmill against overspeeding at a wind velocity higher than the calculated one, with which the maximum

rotation frequency is often limited only by conditions of the strength of the windmill and drive parts. For the synchronized generator, it is necessary to provide the least unevenness in rotation frequency. When working with a centrifugal pump at high values for windmill rotation frequency, cavitation rates are dangerous.

While in the windmill with rotating blades the initial moment can be exceeded by turning the blades to the optimal starting angles, in a windmill with rigidly fastened blades, the pickup, i.e. its capability to accelerate after starting, which depends on moment $\bar{M}_0$, the curvature of characteristic $\bar{M}_{mo}(Z)$ and its shape in the initial section, can be improved within small limits by increasing the width and setting angles of the blades. An increase in the value of $\bar{M}_0$ is also enhanced by improving the load-bearing properties of the blade at stalling flow-past rates which correspond to the initial sections of characteristic $\bar{M}_{mo}(Z)$. This can be achieved by deflecting the nose part of the profile away from the chord or by using a slat.

Using rigidly fastened blades, for better operation of the tail, attempts are made to increase within rational limits the radius of the root section of the blade or the hub r_0. In low-speed windmills, $\bar{r}_0 = r_0/R = 0.35\text{-}0.4$, and in high speed ones, it does not usually exceed 0.25. An increase in r_0 increases the losses of the windmill, but improves the working conditions of the tail. /200

From the expression

$$\bar{T}_{w\Sigma} = \bar{T}_{w\varphi} + \bar{T}_{w.t} = \bar{r}_0^2 + \frac{\xi_1 \eta_1^2}{2Z^2} \ln \frac{1}{r_0}, \qquad (165)$$

where ξ_1 is the coefficient of utilization of the wind energy by an ideal windmill; η_1 is the average value of the windmill's relative coefficient of efficiency for its entire radius; it is possible to determine losses $T_{w0}(\bar{r}_0)$ from underutilization of the area swept by the windmill and losses $T_{wt}(\bar{r}_0)$ due to twisting of the stream behind it. When selecting aerodynamic parameters, it should be considered that with $2.5 \leq Z_n \leq 6.5$, the synchronous speed is usually 2-2.2 times higher than normal and

$$\bar{M}_{mo.n} \approx 5.5 \cdot 10^{-2} \frac{Z_n^2 + 1.36}{Z_n^2}. \qquad (166)$$

The load-bearing frame to which the blades are fastened has a great effect on the characteristics of low-speed windmills. If it is located in front of the blade pattern, the losses in power

in the machine are reduced by 50-60% by comparison with the normally used layout. At the same time, the torque moment of the windmill increases during the nominal and initial rates by 8-10%, and at rates close to the synchronous one, by 20-30%. M. O. Frankfurt [37] proposed an approximate interpolation formula for determining the adjusted coefficient of loading C_y which is constant along the length of the blade of a low-speed windmill.

$$\bar{C}_y \approx a\,(1 + \tau_{0.8})\,\varphi_{0.8}, \tag{167}$$

where a is the coefficient (a $\sim$ 0.022 for a twisted blade and a $\sim$ 0.019 for an untwisted one); $\tau_{0.8} = \dfrac{\overline{ib}_{0.8}}{2\pi\bar{r}_{0.8}}$ is the density at which the blades fill the surface swept by the windmill in a section having a radius $\bar{r} = 0.8$.

$$\varphi_{0.8} = \varphi_0 \frac{0.8 - \bar{r}_0}{1 - \bar{r}_0}(\varphi_0 - \varphi_R); \quad \tau_{0.8} = \frac{i}{1.6\pi}\left[\bar{b}_0 + \frac{0.8 - \bar{r}_0}{1 - \bar{r}_0}(\bar{b}_R - \bar{b}_0)\right].$$

where ϕ_0, $\phi_{0.8}$, ϕ_R are the angle of blade chord in corresponding sections; $\bar{b}_0$ and $\bar{b}_R$ are the relative chord of the blade in the corresponding sections.

Setting the sizes of values from (167) into the formula for initial moment, we obtain

$$\bar{M}_0 = a\,(1 + \tau_{0.8})\,\varphi_{0.8}\,\frac{\tau_0}{3}\,\bar{r}_0(1 - \bar{r}_0)\left[3\,(1 + \bar{r}_0) + \left(\frac{b_R}{b_0} - 1\right)(2 + \bar{r}_0)\right], \tag{168}$$

where τ_0 is the density of blade pattern coverage in a section with radius r_0.

Until recent times, there was a widespread opinion that one of the main factors determining the type of windmill is, together with the load, the average periodic wind velocity. In the meantime investigations have shown that even in areas where velocity $v_{per} \leq 3.5$ m/sec, high-speed machines can be effectively used. At the same time, with the above calculation, design and high quality of manufacturing, a low-speed machine also operates reliability during a strong wind.

The external characteristics of the working machine $\bar{M}_r(\omega)$ and the initial moment of loading resistance M_{0r} must be considered along with the other factors. If the pump is a low-speed

type, having high values for moment M_{0r} and grad M_r during the acceleration rates, a multibladed windmill should be used. With this, diameter $D_{wm\ min}$ is decreased, the drive is simplified and its coefficient of efficiency is increased. If moment M_{0r} is relatively high, but the pump is a high speed type or its moment of resistance increases smoothly, windmills having higher speeds can be used, but the machine must have a relieving system which works in the starting and acceleration rates. In regions with a strong and gusty wind, motors with regulation by means of directing the windmill out of the wind, especially with eccentric placement of the windmill's axis of rotation relative to the axis of rotation of the head, work less reliably and higher gyroscopic loads arise. When computing the load-bearing elements of the windmill, large strength reserves, which increases metal consumption and this also increases gyroscopic loads, must be introduced.

When combining a wind motor with screw-type or pneumatic pumps, windmills of various types can be used. In regions where there is often a wind with high velocities, use of a windmill with rotating blades increases the period of utilization of the machine, since the range of working wind velocities for centrifugal-aerodynamic systems of regulation is usually equal to 4-40 m/sec, and for windmills regulated by directing them out of the wind, it is usually 4-20 m/sec.

Calculations have shown that with the operation of a compressor without a relieving system using a windmill (Z_n = 4.5; D_{wm} = 4.5 m), having three blades which automatically turn to optimal starting angles, the output in zones with a wind velocity v_{per} = 5 m/sec comprises 43% of the total maximum possible output with work in all rates with a maximum coefficient of energy utilization. /202

A two-bladed windmill (Z_n = 5.5; $\phi_n = \phi_0$) in the presence of a compressor relieving system provides an output which is 19% higher than a three-bladed one. The use of a low-speed windmill (i = 18; Z_n = 1.2) decreases output by 4%, makes the machine more complex and increases its weight. The use of two- or three-bladed windmills whose blades are automatically rotated to the most advantageous starting angles provides the best results in machines equipped with automatic relieving systems.

In cases where the use of a windmill with rigidly fastened blades is justified, the problem of improving the starting characteristics and acceleration of the windmill, i.e. its capability to accelerate after beginning rotation, arises in front of the designer. With a set moment $M_r(\omega)$ the run-up of the windmill depends on i_{mach}, the shape of characteristic $\overline{M}_{mo}(Z)$ in its initial section, particularly on the value of $\frac{\partial \overline{M}_{mo}}{\partial Z}$; and on the absolute value for moment $\overline{M}_{mo}$ at low speed.

A number of methods for increasing acceleration are known: increasing the width of the blades and the angle of their inclination, asymmetrical placement of the blades along the swept surface, installation of slats, and deflection of the nose part of the blade profile away from the chord. Investigations of TsAGI showed that for blades with profiles of constant thickness ($\bar{b} = 0.16$) a relatively low overall angle of inclination, i.e. the difference in the angles between the root and tip sections of the blade ($\Delta\phi_{0a} \approx 9°$) and with an increased hub diameter ($\bar{r}_0 = 0.27$, when $\bar{r}_0 = 0.18$-0.22 is usually taken), it is advisable to deflect the nose part of the profile. With this, the speed Z_n decreases little, while at the same time moment $\bar{M}_{mo}$ increases by 30-40%, and moment $\bar{M}_{z_n}$ increases by 12-15%. Thus, in a six-bladed windmill, moment $\bar{M}_0$ increases by almost 50% and moment $\bar{M}_{mo}$, at all values of speed Z in the left portion of the characteristic, by 25-40%. Other methods are less effective or unacceptable for operational considerations.

During recent years, low-speed windmills have, as a rule, been used in machines with a piston pump, and more rarely with a belt water lift. The windmill overcomes the maximum moment of the pump's resistance, which is almost three times that of the average resistance. This lengthens the initial rates and requires an increase in wind velocities, especially because the kinetic energy moving the parts of the machine during the acceleration period is relatively small. With an increase in the number of blades, the effect of wind velocity on the value of the moments and on frontal pressure of the windmill becomes more significant, which can lead to an increase in overloads of the windmill during gusts. /203

During calculation of starting rate according to the average moment of resistance, we have

$$M_{r.av} = \pi R^3 \frac{\rho}{2} \bar{M}_0 v_0^2 \text{ or } v_0 = \sqrt{\frac{2M_{r.av}}{\pi R^3 \rho \bar{M}_0}}$$

according to the maximum moment of resistance, which is π times greater than the average

$$v_0' = \sqrt{\frac{2\pi M_{r.av}}{\pi R^3 \rho \bar{M}_0}} \text{ and } \frac{v_0'}{v_0} = \sqrt{\pi} = 1{,}77.$$

§ 3. Calculation for Strength and Designing Windmills

After the type of windmill has been selected, aerodynamic calculations performed and the parameters of the machine optimized, calculation for the strength and design of the blades and the entire windmill is a no less difficult and vital task.

When designing a low-speed windmill, the section of its elements is usually selected out of design considerations, and static calculations, the majority of which are checking ones, are made according to the maximum forces from frontal pressure of the flow $P_{wm\ max}$ and gyroscopic moments M_g:

$$P_{wm\ max} = C_{y\ max} S_{\Sigma} \frac{\rho}{2} (\omega^2_{2m\ max} r^2_m + v^2_{max}) n; \qquad (169)$$

$$M_{g\ max} = J_{wm} \omega_{wm\ max} \omega_{2\ max}, \qquad (170)$$

where $C_{y\ max} = 1.2$ is the maximum value for the coefficient of lifting force; S_{Σ} is the total area of all blades; v_{max} is the maximum gust velocity; $\omega_{wm\ max}$ is the angular velocity of the windmill with consideration for residual unevenness in regulation; $n \approx 1.3$ is the coefficient of overload; J_{wm} is the moment of inertia of the windmill relative to its axis of rotation; $\omega_{2\ max}$ is the maximum velocity for turning the windmill and the head relative to the vertical axis (see § 4 of this chapter).

The moment $M_{g\ max}$ acts on the main shaft of the wind motor, and therefore it must be considered when calculating the shaft for strength.

During regulation by moving the windmill out of the wind

$$\omega_{2\ max} = \omega_{reg\ max} + \omega_{or\ max}, \qquad (171)$$

where $\omega_{reg\ max}$ is the turning velocity of the plane of rotation of the windmill relative to the vertical axis of the head during the regulation process; $\omega_{or\ max}$ is the turning velocity of the windmill /204 together with the head during the process of orientation according to the direction of the flow.

From Fig. 119 it is evident that combination of the velocities will lead to an essential increase in the total turning velocity of the head and, as a result, to an increase in moment M_g. Therefore, the use of a wind rose system of orientation is also advantageous for motors regulated by turning the windmill away from the wind.

The low-speed windmill of the cantilever type (Fig. 120) is most often used. A windmill of this type consists of a cast hub 2, outer and inner rings of pipe or angle steel, which are connected to each other and to the hub with a system of radial braces:

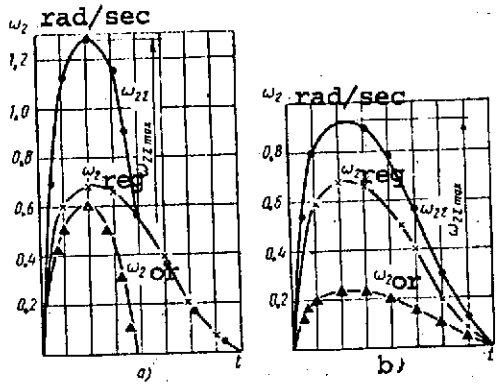


Fig. 119. Determining the maximum turning velocities of the head: a. with a tail; b. with a wind rose orientation system; $\omega_{2\ or}$ -- velocity of reorientation; $\omega_{2\ reg}$ -- velocity during the regulation process; $\omega_{2\Sigma}$ -- total angular turning velocity of the head.

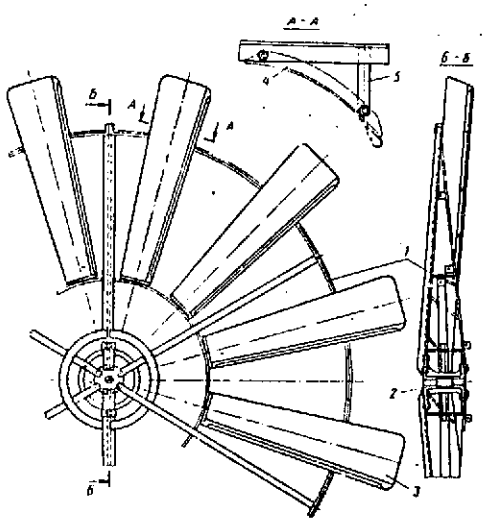


Fig. 120. Design of a low-speed windmill of the bracket type: 1. frame; 2. hub; 3. blade; 4. angle piece; 5. bracket.

front and rear. To give the windmill strength, the braces are connected with crosspieces. The blade 3 usually has two formed parts made of angle steel, which are installed in front of the frame and fastened to it. For convenience in transportation, the frame is often made to be knocked down into two parts. Investigations in recent years have shown that fastening the blades behind the frame (see Fig. 48) improves the characteristics of the windmill, and in particular increases moment M_0.

The blade is manufactured of high quality cold-worked steel with a thickness of 0.8-1.2 mm. For greater strength, the entry edge of the blade is rolled in or given a drop-shaped form. Section A-A of the blade is usually the most loaded one, and must be checked for bending from aerodynamic forces acting on the cantilevered part of the blade. /205

Some earlier-produced machines had a windmill of the bicycle type (Fig. 121), which consists of a rigid rim 1, concentric relative to hub 2 with a system of thin spokes 3 in tension, on which are fastened blades 4. This windmill is lighter than the normal one, but as extened operations showed, it is less reliable. Besides this, the labor consumption for assembling a windmill of the bicycle type is higher than for a normal one.

The use of the rim with spokes justifies itself in, for instance, portable installations, where the radial loads dominate axial ones and the windmill is made light and strong. The use of a bicycle type windmill is not very well justified, since it works in conditions in which the main effects are axial aerodynamic loads and

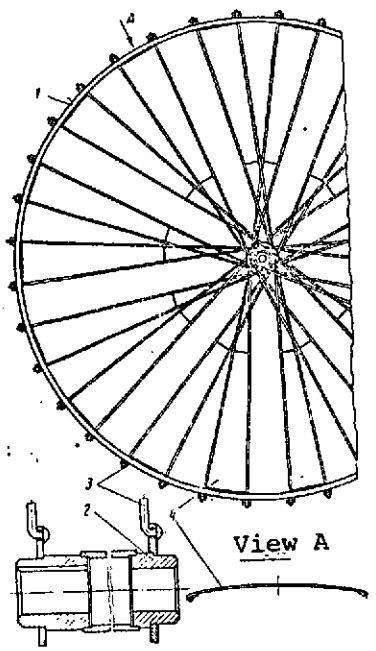


Fig. 121. Low-speed windmill of the bicycle type.

turning-out (bending) moments from forces of gyroscopic origin; tension efforts from centrifugal forces are small, and there are simply no compression radial loads. Therefore, a windmill of this type can be recommended only in special conditions and only for motors having low power ($D_{wm} < 1.5$ m), designed for regions not having strong winds. /206

During the design of a high-speed windmill, the main task consists of providing dynamic strength reliability of the blade. The solution of this problem is made more complex by insufficient knowledge of actual loads. One of the singularities of a blade consists of the fact that an increase in its metal consumption frequently not only does not increase reliability, but, conversely, enhances an increase in the stresses, since a significant part of the changing loads in the blade spar are caused by the weight of the airfoil, and also by gyroscopic forces, which are proportional to the moment of inertia of the blade and connected with its weight. The portion of variable loads in the total stresses increases with an increase in speed Z_n. Thus, in the D-12 motor, bending stresses in the dangerous section of the spar $\sigma_{bend\ max}$ from the weight of the blades with it in the horizontal position comprise up to 50% of σ_Σ and exceed stresses from moment M_{mo} by almost five times. Therefore, a reduction in blade weight is one of the most important problems.

The system and mechanism of orientation have an effect on the value of some loads. Thus, for instance, constant oscillation of the head relative to its vertical axis, occurring with low amplitudes with a high frequency are a source of additional gyroscopic loads. Changes in bending and rotational moments and their periodic pulsations are seen on an oscillogram of the work of a D-12 motor (Fig. 122), and asymmetry of the loading cycle, which should be considered during the selection of admissible loads, is clearly manifested. Experiments in wedging the head showed that gyroscopic loads on the blade decreased by almost 3.5 times. We will note that the existing opinion that the maximum danger for a windmill is storm wind velocities is not always justified, since the windmill head is more stable and gyroscopic loads are reduced at high wind velocities. /207

Designs for high-speed windmills with rotating blades are shown in Figs. 44, 60 and 70. The design of an all-metal blade is

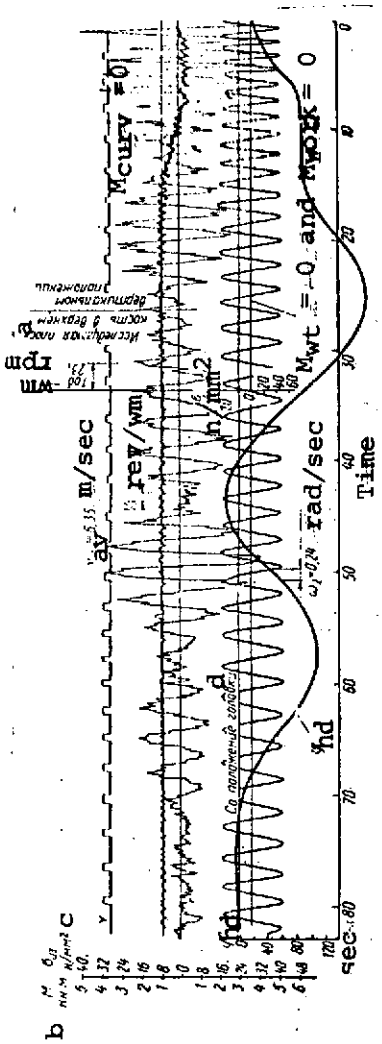


Fig. 122. Oscillogram of changes in moments of wind machine.

Key: a. Investigated plane in upper vertical position; b. $M_{k.nom}$; c. n/mm^2; d. median position of head.

shown in Fig. 123. The blade has a major load-bearing element -- stringer 1, which is made of tubing or sheet, bent into the shape of a letter П in sections. The stringer is ended with spar 2, which is fastened in the hub. The parts of the regulator are in turn fastened to the spar. Ribs 3 with the proper shape are placed along the length of the stringer, and the skin, made of cold-drawn or hot-rolled steel is fastened to them by spot welding. The thickness of the skin σ_{sk} for a windmill having a diameter of D_{wm} = 3-4 m is 0.6-0.8 mm, and the blade length l_b = = 5-6 m, σ_{sk} = 1.2-1.5 mm. For short blades (l_b = 1-2 m) the load-bearing properties of the skin are not usually considered, which increases its strength reserve. The design of the metal blade for a "Sokol" machine differs with its non-round spar with its moment of section inertia which changes according to length. When using pressed fiberglass blades, a universal tip is used instead of a spar. /208

For high-speed windmills with blades made of sheet steel, a high total angle of torsion of blade sections is undesirable. It can be decreased by increasing the relative width $\bar{b}$ of the profile in the peripheral sections and, conversely, by decreasing width $\bar{b}$ in the root sections. In this case, with maintenance of the calculated values for coefficient C_y, the angles of attack of the peripheral sections are decreased, and those of the tips are increased.

As experience shows, the dangerous blade section is the place where the spar is fastened to the hub. Calculations for strength a re made first for this section. A high-speed blade experience up to 10 million loading cycles in its plane of rotation over the course of 2-3 months' work. During design, it is necessary to pay a great deal of attention to machining fineness of the spar surfaces, hermetization of skins and the methods of its fastening. The use of circular welded seams in the dangerous spar section, and stress concentrators -- threads, holes, pressed fittings, etc. -- are, for instance, inadmissible. At the same time, the reliability of the spar is sharply increased with riveting, grinding, and the use of bolted flanged connections.

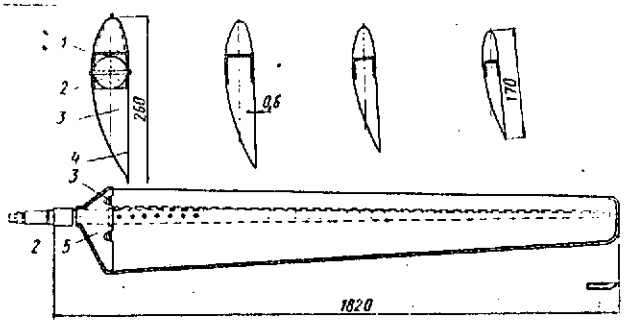


Fig. 123. Design of a metal blade for a D-4 high-speed machine: 1. stringer; 2. spar; 3. rib; 4. skin; 5. fairing.

The ingress of moisture does not always cause rusting of the skin and its intensive destruction, but freezing of the water inside the blade can cause a weight imbalance in the windmill and an increase in vibration. During the design, the possibility for destruction of the weight balance of the windmill must be reduced to the minimum.

Changes in loads on the blade are caused by change in wind rates, unevenness in the field of flow velocities across the swept surface and its acceleration during a gust, the transfer processes of regulation and orientation, centrifugal forces putting the blade in tension, and also its unloading from bending and others. /209

We will examine blade flexion in two planes: in the plane of action of the flow, i.e. in the plane of the least rigidity and in the plane of rotation of the windmill. Twisting oscillations and stresses in these planes are insignificant, and they can be ignored. The aerodynamic forces can be found by determining the profiles of coefficient C_y and C_x according to polar. Out of other loads, the most essential ones are: in the plane of least rigidity -- hydroscopic and those arising during lateral oscillations of the blade; in the plane of rotation -- those from the weight of the blade and from moment M_{mo}.

One of the conditions for reliable operation of a blade is that not one of the harmonic frequencies of the inherent oscillation of the blade must coincide with the frequencies of external disturbing forces. With this, not all harmonics of external forces should be considered, but only those which are sufficient for creating stresses which are essential in size. It is usually possible to limit oneself to examination of the first two tones.

Resonance phenomena in blades arise when the frequencies of disturbing forces are equal or shorter than the frequency of inherent blade oscillations. As the result of the effect of centrifugal forces, the frequency of inherent blade oscillation depends on the frequency of rotation ω_{wm} and therefore the frequency of oscillation of a rotating blade (dynamic) f_{dyn} is always greater than the frequency for a static f_{st} non-moving blade.

$$f_{dyn} = \sqrt{f_{st}^2 + k_j \omega_{wm}^2}, \qquad (172)$$

where k_j is the coefficient depending on the construction of the blade, the shape and type of its fastening to the hub, and also considering the effect of tone j, for which the frequency is sought.

Dangerous frequencies are equal to $j\omega_{wm}$. Since ω_{wm} = var, the frequency of f_{dyn} constantly changes, although within small limits. Therefore, resonance diagrams must be constructed for the entire range of changes of ω_{wm}, right up to ω_{max} of the machine's idling, with wind velocity of $v_{work\ max}$. The probability for blade resonance due to the effect of pulsation of the stream, as experience shows, is small. For powerful machines, the probability of resonance is even smaller due to the large shift in phase and time of impulses of gusts across the surface swept by the windmill. A decrease in the probability of resonance is also promoted by forces of damping. However, considering the frequency of inherent blade oscillations, the appearance of resonance during passage of the blade in front of the tower or behind it can be expected in some cases.

Using the girder model of a blade with constant distribution of parameters and considering the action of centrifugal force P_{cf}, we write the differential equation of blade bending the plane of the action of the flow: /210

$$\frac{d^2}{dz^2}\left(EJ_b\frac{d^2y}{dz^2}\right)-\frac{d}{dz}\left(P_c\frac{dy}{dz}\right)+m\frac{d^2y}{dt^2}=q_b \qquad (173)$$

where y is the change in the blade section along the flow; z is the ordinate along the blade; m is the mass of the blade attributable to a unit of its length; q_b is the intensity of constant distribution of load from forces of frontal pressure perpendicular to the blade.

Having determined the frequency of inherent blade oscillations either experimentally or by the calculation method, and considering that the dangerous frequencies are proportional to the frequency of rotation of the windmill, using resonance diagrams, we establish unfavorable working rates (Fig. 124) [5, 24, 60].

Aerodynamic calculation allows the nominal value of forces acting on the blade to be determined. Under actual conditions, the values of these forces exceed the calculated ones, due to wind gusts, which causes brief overloads in the blade. The coefficient of overload $k_{ovl} = P_{max}/P_n$ depends on the characteristics of the wind gust, the system of regulation, inertial properties and speed of the regulator.

We will locate the components of total aerodynamic force acting on the wind in the established working rate according to characteristics B(Z) and $\bar{M}_{mo}(Z)$.

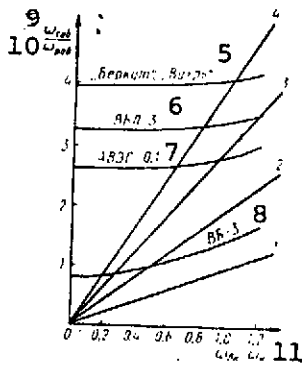


Fig. 124. Resonance diagram of low-power windmill blades: 1-4. serial number of harmonics of inherent oscillation frequencies; 5. "Berkut;" "Vikhr';" 6. VBL-3; 7. AVES-0.1; 8. VB-3; 9. ω_{inh}; 10. ω_{work}; 11. ω_{wm}/ω_n

The force of frontal pressure

$$P_b = \frac{B}{i_b}\pi R^2 \frac{\rho v^2}{2}, \qquad (174)$$

and the tangential force

$$T_b \approx \frac{\bar{M}_{mo}}{i_b \bar{r}_e}\pi R^2 \frac{\rho v^2}{2}. \qquad (175)$$

The summary load with the center of effort of the blade at a distance of $\bar{r}_e = 0.7$ from the axis of rotation will be [38]:

$$P_{b\Sigma} = \pi R^2 \rho \frac{v^2}{2i_b}\sqrt{B^2 + \left(\frac{\bar{M}_{mo}}{\bar{r}_e}\right)^2} \approx \approx 2R^2 v^2 \frac{B}{i_b}\sqrt{1 + 2\left(\frac{\bar{M}_{mo}}{B}\right)^2}. \qquad (176)$$

At low values for Z, the center of effort during a gust is displaced toward the hub by $(0.05\text{-}0.08)r_e$. The summary load during a wind gust $K_{gust} = \frac{v_{gust}}{v_e}$, if the changes in force are considered quasi-stationary, will be /211

$$P_{b\Sigma} = \pi R^2 \frac{\rho v^2}{2i_b} K_{gust}^2 \sqrt{B_{gust}^2 + \left(\frac{\bar{M}_{gust}}{\bar{r}_e}\right)}. \qquad (177)$$

The coefficient of overload from the action of frontal pressure with this is expressed thusly:

$$k_{ovl} = K_{gust}^2 \frac{B_{gust}}{B}\sqrt{\frac{1 + 2\left(\frac{\bar{M}_{gust}}{B_{gust}}\right)}{1 + 2\left(\frac{\bar{M}_{mo}}{B}\right)}}. \qquad (178)$$

The coefficient of summary overload from the action of all aerodynamic forces after simplification and some allowance [37] is

$$k_{ovl\,\Sigma} \approx 0{,}3\bar{i}\bar{b}_0\left(C_{e.1}+\frac{dC_y}{d\alpha}\Delta\alpha_{gust}\right)\times$$
$$\times\left\{\left[\frac{4-\left(1-\frac{b_R}{b_0}\right)\left[3-\bar{r}_0(1+\bar{r}_0)\right]}{12(1-\bar{r}_0^2)}\right]Z^2+\left[\frac{1-\frac{1}{2}\left(1-\frac{b_R}{b_0}\right)}{1+\bar{r}_0}\right]K_{gust}^2\right\}\times \tag{179}$$
$$\times\left[\frac{4K_{gust}^2-2\left(1+\frac{dC_y}{d\alpha}\cdot\frac{\Delta\alpha_{gust}}{C_{e.1}}\right)}{4K_{gust}^2-\left(1+\frac{dC_y}{d\alpha}\cdot\frac{\Delta\alpha_{gust}}{C_{e.1}}\right)}\right],$$

where $C_{e.1}$ is the coefficient of lifting force with the nominal number of moduluses Z_n; $\Delta\alpha_{gust}$ is the change in the angle of attack during a gust.

For a windmill with $Z_n = 5$ at $K_{gust} = 1.3\text{-}1.6$, we obtain $k_{ovl\,\Sigma} = 1.4\text{-}2$ respectively. In certain cases, the overload may be four to five times.

During calculation for strength, it is possible to accept a linear distribution along a triangle of aerodynamic loads without considering tip losses and the initial radius of the blade $\bar{r}_0$ (Fig. 125). Having determined forces $P_{wm.aer}$ and $P_{b.aer}$ and multiplying them by the coefficient of overload k_{ovl}, we locate the moment from the action of force $P_{b.aer}$, bending the spar,

$$M_{b.aer} = (r_{c.e} - r_0)P_{b.aer},$$

where $r_{c.e}$ is the distance from the axis of rotation of the windmill to the center of effort of the airfoil; the rotational and gyroscopic moments

$$M_{mo} = \frac{N_{wm}}{i_b\omega_{wm}}\,k_{ovl}; \tag{180}$$

$$M_g = 2J_b\omega_{wm}\omega_2\sin\omega_{wm}t. \tag{181}$$

For a blade with mass m_b, centrifugal forces are

$$P_{c.f} = m_b\omega_{wm}^2 r_{c.e}; \tag{182}$$

and the moment from the action of weight is

$$M_{wt} = G_b(r_{c.e} - r_0). \tag{183}$$

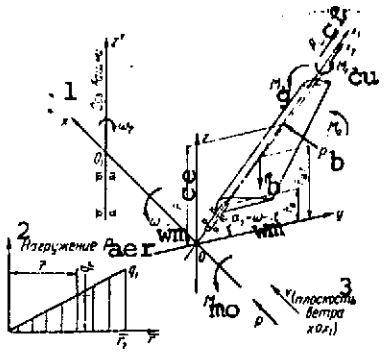


Fig. 125. Diagram of blade loading.

Key: 1. Axis of tower; 2. Loading P_{aer}; 3. (plane of wind xOx_1)

The moment of centrifugal forces relative to the longitudinal axis of rotation of the blade is

$$M'_{cf} = J_b \frac{\omega_{wm}^2 \sin 2\phi}{2} \qquad (184)$$

Since the center of effort does not lie on this axis, moment $M_{b.aer}$ from the total aerodynamic force P_{aer} will be equal to $P_{aer} l_a$.

With consideration for the coefficient of dynamics for the s-th harmonic of disturbing force (according to the data of V. F. Mosalev)

$$k_s = \frac{1}{1 - s^2\zeta^2}, \qquad (185)$$

where s is the serial number of the disturbing force harmonic; $\zeta = \nu/f$ is the relationship of the frequency of the disturbing force to the frequency of the inherent oscillation of the system, we obtain the maximum load

$$P_{b\ max} = P_b k_{ovl} + k_s P_s, \qquad (186)$$

where P_s is the variable part of the load, caused by gyroscopic forces and other factors [5, 23].

The coefficient of loading for the blade by centrifugal forces in the section at distance r_0 will be

$$\beta_0 = \frac{1}{1 + \frac{\omega_{wm}^2}{l_{dyn}}}. \qquad (187)$$

At a distance of $\bar{r} = 0.2$ for the VB-3 machine, the coefficient $\beta_0 = 0.6$; and for the BVL-3 and AVES-0.1, $\beta_0 = 0.9$.

In the final form we obtain the following expression for determining moments and forces:

$$M_{aer}(r) = k_m k_d M_{r=0} \beta_0(r); \tag{188}$$

$$M_g(r) = 2J_b(r)\omega_{wm}\omega_2 k_s \beta_g(r) \sin \omega_{wm} t; \tag{189}$$

$$M_{b.wt}(r) = G_b k_s' k_1 \beta_{\omega_{wm}}(r) \cos \omega_{wm} t \int_{r_i}^{R} rm(r)dr; \tag{190}$$

$$P_{cf}(r) = k_{cf}\omega_{wm}^2 \int_{r_0}^{R} rm(r)dr; \tag{191}$$

$$M_\Sigma(r) = \sqrt{[M_{aer}(r) + M_g(r)]^2 + [M_{mo}(r) + M_{wt}(r)]^2}, \tag{192}$$

where k_m is the coefficient of change of the aerodynamic bending moment along the length of the blade; k_d is the coefficient of dynamics of overload of the blade; k_s is the coefficient of dynamics of the blade for the s-th harmonic of disturbing force; k_s' is the coefficient of dynamics of the blade in the plane of rotation of the windmill; k_1 is the coefficient of blade overload in the plane of rotation of the windmill; $\beta_{\omega_{wm}}$ is the coefficient of unloading of the windmill in the plane of rotation; k_{cf} is the coefficient of centrifugal force increase.

The summary stresses in section r from the action of all forces and moments will be

$$\sigma_\Sigma(r) = \sigma_{cf} + \sqrt{(\sigma_{aer} + \sigma_g)^2 + (\sigma_{mo} + \sigma_{wt})^2} \tag{193}$$

Reserves in strength according to σ_{max} and amplitude σ_a of the cycle are

$$n_{max} = \frac{\sigma_y \epsilon_y}{\sigma_{max}}; \quad n_a = \frac{\sigma_{-1}}{(\sigma_a)_e} \tag{194}$$

where σ_y is the yield point; σ_{-1} is the fatigue point with symmetrical cycle; ϵ_y is the coefficient of change in stresses due to the effect of the absolute dimensions of the parts; σ_{max} is the maximum stresses of asymmetrical cycles; $(\sigma_a)_e = (k_\sigma)_D \sigma_a + \psi_\sigma \sigma_m$ is the stresses of symmetrical cycle, equivalent

to given stress of asymmetrical cycle and reflecting the effect of design and engineering factors on fatigue resistance; σ_m is the median stress of cycle; ψ_σ is the coefficient of shape of the diagram of limiting stresses; $(k_\sigma)_D$ is the coefficient of stress concentration.

The reserves in strength with the blade in the horizontal position are: according to maximum stresses /214

$$n_{max} = \frac{\sigma_y \varepsilon_y}{\sigma_{cf} + \sqrt{(\sigma_{v_m} + \sigma_{v_a})^2 + (\sigma_{wt} + \sigma_{mo})^2}}, \qquad (195)$$

where σ_{vm} and σ_{va} are the stresses from the effect of moments M_g and M_{aer} respectively in the plane of action of the airstream;

According to stress amplitude

$$n_a = \frac{\sigma_{-1}}{(k_\sigma)_D \sigma_a + \psi_\sigma \sigma_m} = \frac{\sigma_{-1}}{(k_\sigma)_D \sqrt{\sigma_{v_a}^2 + \sigma_{wt}^2} + \psi_m \left(\sigma_{cf} + \sqrt{\sigma_{v_m}^2 + \sigma_{\omega_m}^2}\right)}, \qquad (196)$$

where $\sigma_{\omega m}$ is the stresses from the action of moment M_r in the plane of rotation of the windmill.

It is recommended that $n_{max} = 1.3\text{-}1.4$ and $n_a = 1.2$ and 1.3 be used.

With the blade in the vertical position, calculations are conducted according to the same formula, but the following are used:

$$\sigma_a = \sigma_g$$

$$\sigma_{max} = \sigma_{cf} + \sqrt{(\sigma_{v_m} + \sigma_g)^2 + \sigma_{mo}^2} \qquad (197)$$

$$\sigma_m = \sigma_{cf} + \sqrt{\sigma_{v_m}^2 + \sigma_{mo}^2} \qquad (198)$$

where σ_{vm} is the median stress of the cycle in the plane of action of the flow; σ_m is the median stress of the cycle.

In conclusion, we will examine calculations for strength not according to nominal load P_n, which would lead to unjustified expenditures in metal, but according to median adjusted (equivalent) load P_{eq}, whose action over the period of operation of the

machine is equal to the summary actionof variable loads P_1, P_2, ..., P_k, causing stresses σ_1, σ_2, ..., σ_k, whose number of applications comprises $z_{\sigma 1} = k_1 t_1 \omega_1$; $z_{\sigma 2} = k_2 t_2 \omega_2$, ..., $z_{\sigma k} = k_k t_k \omega_k$.

$$P_{eq} = P_1 \sqrt[m_1]{\sum \left[\frac{z_{\sigma i}}{\sum z_{\sigma i}} \left(\frac{P_i}{P_1} \right)^{m_1} \right]}, \tag{199}$$

where m_1 is the index of the degree of Weller endurance curve ($\sigma_i^{m_1} N_i = \text{const}$); N_i is the number of stress cycles of the element.

It is important to pay attention to the fact that in expression (199), only those members causing overstressing, i.e. having higher values than the extended fatigue limit σ_{r_k}, are subjected to summarization.

The calculation according to equivalent loads ($P_{eq} < P_{max}$) is a calculation for the limiting longevity and is actual under conditions $\Sigma z_{\sigma i} \leq N_0$ (where N_0 is the base number of cycles corresponding to σ_{r_k}); in this case, the calculated dependency has the appearance /215

$$\sigma_{eq} = \sigma_1 \sqrt[m_1]{\sum \frac{z_{\sigma i}}{\sum z_{\sigma i}} \left(\frac{\sigma_i}{\sigma_1} \right)^{m_1}} \leq \sigma_{r_{adm}} \sqrt[m_1]{\frac{N_0}{\sum z_{\sigma_0}}}, \tag{200}$$

where σ_{eq} and σ_n are the stresses corresponding to the equivalent P_{eq} and nominal P_n loads.

Then the calculated service life will be

$$L = \frac{N_0}{\sum z_{\sigma i} \left(\frac{\sigma_i}{\sigma_{r_k}} \right)^{m_1}}. \tag{201}$$

§ 4. Calculation and Design of Automatic Orientation Mechanisms

The system of automatic orientation essentially influences reliability, output and other operational indices of a machine. Thus, with insufficient sensitivity of the orientation mechanism angle γ_1 between the axis of rotation of the windmill and the airflow velocity vector can prove to be relatively high, which will lead to a decrease in the power of the motor: with $\gamma_1 = 5°$, losses in power comprise nearly 10%. The extended operation of a high-speed windmill in a configuration of oblique blowing is undesirable. At the same time, with a very high velocity of

reorientation, gyroscopic loads increase, which necessitates reinforcement of the blades, the main shaft and other elements.

Selection of the orientation system is affected by the power of the machine, the type and speed of the windmill, its placement relative to the tower, the system for regulating rotation frequency, etc. The basic requirements outline for an orientation mechanism are the following:

1) With $v = v_0$, precision of orientation, i.e. maximum allowable value of γ_1, must be within limits of 4-5° (maximum permissible reduction in moment $\bar{M}_0$ is 3%), and with $v = v_{n.reg}$ and higher, $\gamma_1 \leq 2\text{-}3°$;

2) The maximum velocity ω_2 of reorientation must not exceed the calculated one, which is usually equal to 0.4-0.5 rad/sec;

3) In all working rates of the machine, stability of the head in the airflow must be assured.

One of two main methods for orientation are usually adopted: using a tail empennage (Fig. 126, a) or using lateral rotating vanes (wind rose), located in front of the windmill (Fig. 126, b) or behind it (Fig. 126, c).

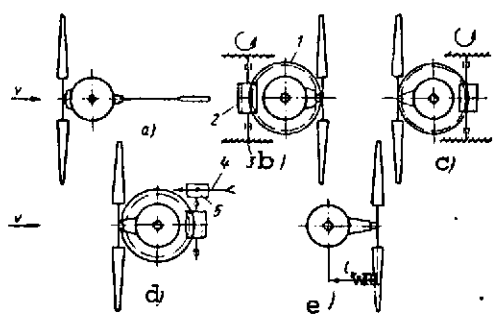


Fig. 126. Systems of automatic orientation: a. using a tail; b and c. using a wind rose located in front of the windmill and behind the windmill; d. using a carrying drive device; e. with the windmill face behind the tower; 1. worm wheel on the the tower; 2. worm gear; 3. wind rose; 4. vane; 5. electric motor.

The tail empennage acts /216 similar to a wind vane: when the wind is directed perpendicular to the plane of rotation of the windmill, no forces deviating in any direction act on the empennage of the tail. With a change in the vector of wind velocity, force P_{ta} arises on the tail and rotates the tail with the head relative to the vertical axis until the axis of the windmill is parallel to the axis of the flow.

The use of wind roses is based on the fact that while the wind velocity vector is parallel to the plane of their rotation and the axis of the windmill, they are stationary. With a change in the direction of the wind, the wind roses begin to rotate. The wind roses are connected with a drive, one of the lengths of which is rigidly fastened to the tower. As a result of this,

rolling (rotation) of the head relative to the tower occurs until the wind roses stop, i.e. until the plane of their rotation becomes parallel to the flow.

The principle of self-orientation of the head due to the effect of resetting turning moments, which arise during oblique blowing of a windmill located behind the tower (Fig. 126, e) is also used. Finally, in large machines, electrical drive devices controlled by an indicator of the wind's direction (vane) is used. In principal layout, this drive mechanism (Fig. 126, d) is similar to the wind rose (Table 25).

When calculating orientation systems, it is necessary to determine the forces and moments acting on the head, tail or wind roses with the windmill working. With oblique blowing of the windmill, the structure of the flow in the zone of the tail is changed. It has been established [37] that the initial data for calculating orientation systems with rigidly fastened tails, i.e. /218
when the tail does not change its position relative to the windmill, occurring in high-speed machines and those in motors regulated by rotating the windmill are different. When calculating orientation systems for low-speed windmills, it is necessary to consider not only braking, but also skewing of the stream, which affects the operation of the tail in the zone where the field of velocities is deformed.

We will assume that the relative moment $\bar{M}_\gamma$ and the coefficient of frontal resistance B_γ during rotation of the windmill change proportional to $\cos^2 \gamma_1$ (where γ_1 is the angle of rotation of the head relative to the direction of the flow), and the value for Z_γ is proportional to $\cos \gamma_1$. With oblique blowing of the windmill, the braking of the flow is decreased, and its minimum velocity, averaged across the entire zone of the tail, is equal to

$$\tilde{v}_{ta} = 1 - k \frac{2e}{1 + e} \cos \gamma_1, \qquad (202)$$

where k is the coefficient considering braking of the stream from the body of the windmill ($k \approx 1.15$).

In a zone of the tail behind a low-speed windmill, velocities v_{ta} can decrease by 25-50% by comparison with the velocity of the initial stream. We will determine angle δ_k of the skew of the flow according to approximation formula

$$\delta_\kappa \approx 28{,}7 B_{\gamma_0} \sin \gamma_1 \cos^2 \gamma_1 \left[1 - \left(\frac{\bar{l}_{\kappa\theta} \sin \gamma_1}{\bar{R}_\partial - \bar{a} \sin \gamma_1}\right)^2\right]^2, \qquad (203)$$

TABLE 25. BASIC METHODS OF AUTOMATIC ORIENTATION

Method	Diagram	Advantages	Deficiencies	Where used
Using a tail empennage	Fig. 126, a	High precision of orientation; simplicity in design of mechanism; relieving tower from twisting moments	High velocity of reorientation and mechanism weight; balancing of head made difficult	1. Low-speed motors 2. Low-powered, high-speed motors and motors regulated by turning the windmill out of the wind
Rotating vanes (wind roses), located: behind the windmill in front of the windmill	Fig. 126, c Fig. 126, b	Low metal consumption and compactness of mechanism; low reorientation velocities; for layout in Fig. 126, b -- also high sensitivity	Complication of construction of the machine; loading the tower with the moment of twisting forces	High-speed motors with a power of from 1.5 kW, regulated by rotating the blades
By locating the windmill behind the vertical axis	Fig. 126, e	Absence of special orientation mechanism	Unstable conduct of the head in the flow; high reorientation velocities; poor head centering	Low power motors
With electrical drive device	Fig. 126, d	Low values for reorientation velocity; high orientation precision	Complication of mechanism design; necessity for external power source	High-power wind electric station motors

where $\bar{l}_{ta} = l_{ta}/R$ is the relative length of the tail; $\bar{R}_d = R_d/R$ is the relative radius of a penetrable disk, equivalent to the windmill; $\bar{a} = a/R$ is the relative span of the windmill.

For low-speed motors $\delta_k \leq 8°$ with $\gamma_1 = 30°$.

We determine the moment of the tail according to the expression

$$\bar{M}_{ta} = k_{ta} C_R \bar{F}_{ta} \bar{l}_{ta}, \tag{204}$$

where $\bar{F}_{ta} = \dfrac{F_{ta}}{\pi R^2}$ is the relative area of the tail.

The data in Table 26 can be used for calculating a tail.

TABLE 26. AVERAGED VALUE FOR PARAMETER $\bar{M}_{ta}$ AND k_{ta} OF LOW-SPEED MOTORS [37]

γ_1	Parameter	Values of parameters at γ_2					
		5°	10°	15°	20°	25°	30°
0	$\bar{M}_{ta}$	—	—	0,032	0,035	0,033	0,04
	k_{ta}	0,2	0,34	0,4	0,35	0,27	0,27
15°	$\bar{M}_{ta}$	—	—	0,03	0,03	0,4	0,55
	k_{ta}	0,58	0,42	0,34	0,3	0,33	0,38
30°	$\bar{M}_{ta}$	—	0,014	0,02	0,035	0,065	0,103
	k_{ta}	0,7	0,26	0,12	0,32	0,5	0,7
45°	$\bar{M}_{ta}$	0,013	0,022	0,045	0,078	0,122	0,153
	k_{ta}	0,87	0,43	0,5	0,68	0,87	1,02
60°	$\bar{M}_{ta}$	0,018	0,04	0,07	0,105	0,145	0,155
	k_{ta}	0,9	0,8	0,92	1,05	1,08	1,0

For calculating an orientation system using a tail empennage, it is necessary to know the aerodynamic forces acting on the windmill during oblique blowing, the moment of the forces of friction in the bearings of the head M_{br}, and in the presence of a vertical shaft, also the reaction moment. If regulation of the motor occurs by turning the windmill out of the wind, the tail participates in the regulation process, and it is necessary to consider the forces on the lateral plane, and if regulation occurs through an eccentric, it is necessary to consider the rotational moment of the windmill. All the forces listed are variable and depend on the working rates of the machine and the velocity of the wind.

For compensating the reaction moment M_{ra} and the forces on the lateral plane, the empennage at the tail is installed at some angle to the axis of the windmill. The moments of the tail relative to the axes are equal to /219

$$M_x = Ty; \quad M_y = Pz_1 - Tl_1; \quad M_z = Py. \tag{205}$$

The equilibrium position of the head is determined by the equation

$$M_{ra} - M_y = M_{ta}. \tag{206}$$

When calculating the moment of the forces of friction in the bearings M_{br}, the weight of the head, the effect of frontal pressure, and also component R_x from aerodynamic forces on the tail, acting along the tail and absorbed by the head bearings, are considered.

Having set the required precision of orientation and allowable values for velocity ω_2, the span of the tail l_{ta} [usually $l_{ta} = (1\text{-}1.4)R$] or the form and dimensions of the empennage are selected, with design considerations being kept in mind. A trapezoidal empennage is more advantageous with short lengths.

At small angles γ_2 (up to 15°), it is possible to consider that

$$M_{ta} = P_{ta} l_{ta} = C_R F_{ta} \frac{\rho v^2}{2} \left(\frac{1 - e}{1 + e}\right) l_{ta}, \tag{207}$$

where $(1 - e)/(1 + e)$ is the value considering the loss in flow velocity behind the windmill; $C_R = \sqrt{C_x^2 + C_y^2}$ is the coefficient of equally acting forces.

Without considering the forces of inertia and the forces of friction in the head, we obtain the orienting value of l_{ta} from the condition /220

$$l_{ta} = \frac{v_{max}}{3\omega_{2\ adm}}. \tag{208}$$

For a low-speed machine with $D_{wm} = 3$ m, the parameters of the orientation system are, for instance, the following: $F_{ta} = 0.49$ m^2

($\bar{F}_{ta}$ = 0.07); l_{ta} = 2.5 m ($\bar{l}_{ta}$ = 1.67); empennage length λ_{ta} = = 1.57; and the shape of the empennage is that of a trapezoid. For three-bladed motors with D = 4 m, tails having $\bar{l}_{ta}$ = 1.25-1.4; and F_{ta} = 1.05-1.15 m^2 are used. It should be emphasized that a decrease in the reach of the tail is undesirable, since the angular velocity of reorientation increases and the gyroscopic loads also increase. Conversely, an increase in reach deteriorates working conditions of the head, creating high possibilities for the appearance of vibration. Usually, for design considerations, $\bar{l}_{ta}$ is taken to be 1.3-1.5, and F_{ta} is computed according to the formula

$$F_{ta} = \frac{2(M_{ra} - M_y)}{v\rho_{ta}^2 C_R l_{ta}}. \qquad (209)$$

For increasing the load-bearing properties of the empennages of tails and lateral planes, small plates are installed on the rear edges of empennages of tails having a surface of short length, and are bent toward the working side. This acts as if increasing the maximum concavity of the profile and increases the load-bearing properties by 1.5-2 times. In cases where the dimensions of the empennage turn out to be extremely large, two- or even three-planed empennages are used. This also promotes a reduction in the maximum angular velocity of reorientation. The aerodynamic characteristics of these empennages, and also of windmills during oblique blowing which are necessary for calculations are presented in some works [7, 35, 37].

If the geometric dimensions of the motor and the nominal sizes of values M_n, P_n and Z_n are known with the angle of incidence of the flow $\gamma_1 = 0$, their values with oblique blowing will be

$$M_{mo\ \gamma} = M_{mo.n} \cos^2 \gamma_1; \quad Z_\gamma = Z_n \cos \gamma_1;$$

$$P_{frt\ \gamma} = P_{frt.n} \cos^2 \gamma_1.$$

We note that utilization of the method of V. P. Vetchinkin for calculating the velocity of reorientation usually gives a higher value of ω_2, since angle γ_1 is taken as equal to 90°, which is improbable in practice. Besides this, the damping action of the tail empennage

$$M_{damp} = C_y F_{ta} \frac{\rho(\omega_2 l_{ta})^2}{2} l_{ta} \qquad (210)$$

and the shifting of the point of application of the equally acting aerodynamic forces during oblique blowing of the windmill are not considered.

V. F. Mosalev showed that under actual conditions, with consideration for the entire accumulation of forces and moments applied to a rotating head, values of ω_2 are lower by comparison with those received according to the method of V. P. Vetchinkin: by 3-4 times with $\gamma_1 = 20°$ and by 50-70% with $\gamma_1 = 40°$. /221

For calculation of angular velocities of the rotation of a head with a tail, we write an equation of moments

$$M_x = Tz_{wm}; \quad M_y = P_{aer}z_{wm}; \quad M_z = P_{aer}y_{wm} - Tl_{wm},$$

where y_{wm}, z_{wm} are the coordinates of the center of effort of the windmill; l_{wm} is the span of the windmill.

The moment of the forces of frictions in the head bearings with consideration for the forces of frontal pressure on the windmill and the effects of the gyroscopic moment (see Fig. 130)

$$M_{br} = \left(f_A r_A \frac{h + l_{hi}}{l_{hi}} + f_B r_B \frac{h}{l_{hi}}\right) P_n \cos^2\gamma_1 + (f_A r_A + f_B r_B)\frac{M}{h} g, \qquad (211)$$

where f_A, f_B are the coefficient of friction in the bearings; r_A, r_B are the averaged radii of bearings.

Knowing the reaction moment $M_{ra} = N_{wm}/\omega_{wm}$ and having written the angular velocity of the head $(d^2\gamma_1)/(dt^2) = 1/2 \cdot (d\omega_2^2)/(d\gamma_1)$, we obtain the equation of its motion in the final form

$$\begin{gathered}
\frac{1}{2} J_{hd} \frac{\omega_2^2}{2} = A_1 a_1 - B_1 b_1 - E\left(\frac{\gamma_1}{2} + \frac{\sin 2\gamma_1}{4}\right) - \\
- \gamma_1 (Q\omega_2^2 + U\omega_2 + M_{mb}), \\
A_1 = 0{,}043 P_{fr} R_{fr}, \\
a_1 = \frac{\gamma_1 \sin\gamma_1}{12} + \frac{\cos 3\gamma_1}{36} + \\
+ \frac{3}{4}(\gamma_1 \sin\gamma_1 + \cos\gamma_1) - 0{,}78; \qquad (212) \\
B_1 = \frac{k_{cf}}{2} F \rho l v^2; \\
b_1 = \frac{\gamma_1^2}{2} - 2k_{ta}\frac{\beta}{\pi}\left(\frac{\beta}{\pi}\sin\frac{\pi}{\beta}\gamma_1 - \gamma_1\cos\frac{\pi}{\beta}\gamma_1\right) + \\
+ \frac{k_{ta}^2}{4}\left[\gamma_1^2 - \frac{\beta}{\pi}\gamma_1 \sin\frac{2\pi}{\beta}\gamma_1 - \frac{\beta^2}{2\pi^2}\left(\cos\frac{2\pi}{\beta}\gamma_1 - 1\right)\right]; \\
E = P_{fr}\left(f_A r_A \frac{h + l_{hi}}{l_{hi}} + f_B r_B \frac{h}{l_{hi}}\right);
\end{gathered}$$

$$Q = \frac{C_y}{2} F \rho l^3;$$

$$U = \frac{2J\omega_{wm}}{h}(f_A r_A + f_B r_B).$$

When calculating the wind rose system, the initial data are the adjusted (usually to the shaft of the wind rose) summary moment of resistance, considering friction in the bearings of the head and in the orientation mechanism drives, the required precision of orientation and the maximum allowable reorientation velocity. Sensitivity of the mechanism with set values of v_0 and $\gamma_{1\ min}$ depends on the characteristics of the wind rose (diameter d_w, the number of blades $i_{b.w}$, and the angle of their setting $\phi b.w$, their number i_w and the gear ratio of the mechanism from the shaft of the wind rose to the gear fastened on the tower $i_{m.oa}$. Usually, the designer selects wind roses according to a reference book and compute $i_{m.oa}$

$$i_{m.oa} = f(M_{br\ \Sigma};\ v_0;\ \gamma_{1\ min};\ d_w;\ i_{b.w}). \qquad (213)$$

Knowing the initial moment of the wind rose $\bar{M}_{w0}$ for angle $\gamma_{1\ min}$, we find any parameter from the expression

$$M_{br\ \Sigma} \leq 2.45 \bar{M}_w d_w^3 v_0^2 i_{b.w} i_{m.oa} 10^{-2}. \qquad (214)$$

When calculating moments of resistance in the orientation mechanism, it is necessary to consider that its loading is low, and therefore the wind roses work at rates which are close to synchronous rates, and the coefficient of efficiency of the mechanism will be

$$\eta_{m.o} = \frac{1}{\frac{1}{\eta} + \frac{\alpha_z}{\chi} - \alpha_z}, \qquad (215)$$

where $\chi = P_{act}/P_c$ is the degree of loading of the mechanism; η is the coefficient of efficiency of the mechanism at P_c; α_z is the relationship of the moment of loss at idle to the moment at calculated load ($\alpha_{z_{av}} \approx 0.075$).

During calculations, the coefficient of efficiency of a toother pair is taken as: for a cylindrical one -- 0.97, and for a conic one -- 0.96. For a worm pair, we have

$$\eta_{wp} = \frac{\tan \alpha_w}{\tan (\alpha_w + \rho_w)} \eta_b, \qquad (216)$$

where α_w is the angle of inclination of the average spiral line of the worm; ρ_w is the angle of friction; η_b is the coefficient of efficiency of the bearings.

The angular velocity of head rotation is

$$\omega_2 = \frac{\omega_w}{i_{m.oa}} = \frac{2Z_{w0}v}{i_{m.oa}d_w}, \qquad (217)$$

where ω_{w0} is the angular velocity of the wind rose; Z_{w0} is the synchronous number of moduluses of the wind rose, determined according to experimental characteristics. /223

From here,

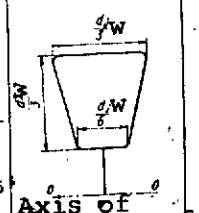
$$i_{m.oa} = \frac{\omega_w}{2} = \frac{2Z_{w0}v_{max}}{d_w\omega_{w\ adm}}. \qquad (218)$$

For the most difficult case, i.e. when the velocity ω_2 can be higher, γ_1 is considered to be 90°, and v_{max} is considered to be 20-25 m/sec. At minimum velocity v_0, the sensitivity of the orientation mechanism is the least, and, since the wind roses at γ_1 = 10-20° are shadowed by the windmill, the value of velocity v_0 must be reduced by 10-15%.

Usually, one or two wind roses with a diameter d_w = (0.15-0.12)D_{wm} with an angle of blade setting ϕ_b = 20-30° are used. The most advantageous are trapezoidal blades, which give a 15-30% greater moment $\bar{M}_{w0}$ than round or rectangular blades which are equal in area. For a wind rose with an angle ϕ_b = 20°, the values of $\bar{M}_{w0}$ and Z_{w0} are presented in Table 27. Full characteristics of wind roses with trapezoidal blades are given in Fig. 127.

Design executions of orientation mechanisms can be the most widely varied. When placing the wind rose in front of the windmill, the most compact mechanism is a worm one, single- or two-staged (Fig. 128 and 66, 78). In the first case, the worm is the wind rose shaft, and the worm gear is colocated with the support bearing of the head. This mechanism, with a gear ratio $i_{n.oa}$ = 100 is used in the "Vikhr'," "Veterok" and other machines,

TABLE 27. CHARACTERISTICS OF A SIX-BLADED WIND ROSE

γ_1	0	30°	45°	60°	75°	85°	Shape of blade
z_{rwO}	3.5	2,85	2.4	1.45	0,6	0.15	
$\overline{M}_{wO}$	0.05	0,046	0,04	0.026	0,015	0.005	Axis of rotation of wind rose

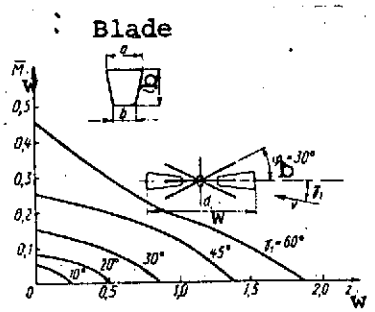


Fig. 127. Characteristics of a six-bladed wind rose: $l_b = a = 0.35d_w$; $w = 0.28d_w$.

providing high sensitivity of the system: with $\gamma_1 = 45°$, $v_{min} = 3.6$ m/sec; with $\gamma_1 = 10°$ and $v = 5.15$ m/sec, the error in head setting with respect to the velocity vector does not exceed 3°. At the same time, experiments have shown that the maximum angular velocity of reorientation does not exceed 0.51 rad/sec (with $\gamma_1 = 45°$ and $v = 21$ m/sec). /22

For larger machines, for instance, the "Sokol," a two-staged mechanism is more advantageous, because it provides the possibility of using small wind roses with high gear ratios ($i_{m.oa}$ = = 1000-2000), and does not separate them from the axis of rotation of the head.

On the contrary with the wind rose placed behind the windmill, they should be removed as far as possible from the axis of rotation of the head. Therefore, a gear mechanism with a so-called pin gear (Fig. 129, a) or combination gear (Fig. 129, b) is used. The advantage of worm pairs is their compactness with high transmission ratios $i_{m.oa}$, and the possibility for reducing clearances in the engagement by means of using a slotted, spring-loaded worm, in which one coil has a smaller pitch and therefore the axial clearance between the teeth of the wheel and the worm is decreased.

Self-orientation of a windmill located behind the tower takes place due to the effect of a moment, whose minimal value must be equal to (see Fig. 126, d)

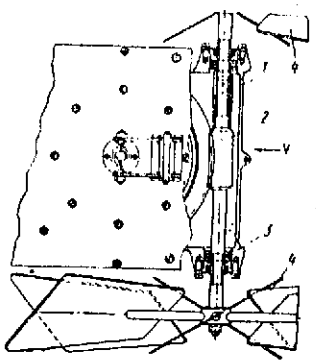


Fig. 128. Design of a wind rose orientation mechanism: 1. worm wheel; 2. worm gear; 3. body of rotating device; 4. wind rose.

$$M_{or} = P_{aer.obl} l_{wm} = \bar{y}\pi R^2 \frac{\rho v^2}{2} l_{wm}, \quad (219)$$

where $P_{aer.obl}$ is the aerodynamic force acting on the windmill with oblique blowing; l_{wm} is the distance from the plane of rotation of the windmill to the axis of rotation of the head; $\bar{y}$ is the coefficient of aerodynamic force. /225

According to the data of V. Ye. Fedotov, with $\gamma_1 \leq 30°$

$$\omega_2 = cRv \sqrt{\frac{l_{wm}}{J_{hd}}} \sqrt{\gamma_{1n}^2 - \gamma_1^2}; \quad (220)$$

$$\omega_{2\max} = cRv\gamma_1 \sqrt{\frac{l_{wm}}{J_{hd}}}, \quad (221)$$

where $J_{hd} = J_{wm} + M_{wm} l_{wm}^2$ is the moment of inertia of the head; m_{wm} is the mass of the windmill; γ_{1i} and γ_1 are the initial and continuing angles of the approach of the airstream to the windmill in rad; $c \approx 0.3$ is a coefficient.

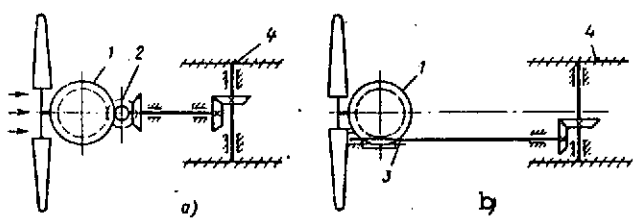


Fig. 129. Diagrams of wind rose orientation mechanisms: a. geared; b. combination; 1. windmill on the tower; 2. conic and cylindrical gear block; 3. worm gear; 4. wind rose.

§ 5. Calculation and Design of Support Devices for Rotating Heads

The supporting device for a rotating head is an important part of the motor, absorbing the major loads. Its design to a significant degree determines the sensitivity of the head to changes in the wind velocity vector, and also its stability in the stream. The major loads on the head and its supporting elements are caused by (Fig. 130):

1) The weight of the windmill and mechanisms composing the head or going into the automatic regulation and orientation systems;

2) Forces of frontal pressure on the head, and in the presence of a lateral plane, also on its vane;

3) The moment of gyroscopic forces acting in the vertical plane, which passes through the axis of the windmill and the axis of rotation of the head;

4) The moment from the weight imbalance of the head relative to its horizontal axis, passing through the axis of rotation of the head. /226

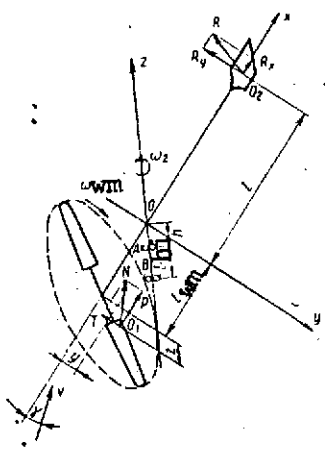


Fig. 130. Loads acting on the rotating head of a wind motor.

The forces, with the exception of the weight of the head, change depending on the velocity of the wind, rotation frequency of the windmill, angular velocity of reorientation, and position of the elements regulating power and windmill rotation frequency, which are in turn determined by the relationship of the moment of moving forces and the moment of load resistance.

For high-speed motors, regulated by rotation of the blade or its tip, the position of the center of gravity of the head practically remains constant; the maximum loads arise during rates corresponding to the maximum calculated wind velocity when the machine is working with the load disconnected and with the maximum angular rotation of the flow velocity vector.

When turning the windmill away from the wind with a lateral plane, the position of the center of gravity of the head changes in the process of regulation. The maximum disequilibrium takes place when the plane and tail are locked, and the plane of rotation of the head coincides with the direction of movement of the flow. However, in this case $P_b = 0$, and $M_g \neq 0$. The maximum loads on the head act with $v = v_{n.reg}$. Calculations for support devices should be made according to these loads.

We will determine frontal pressure on the windmill according to formula (169) or through a coefficient of frontal pressure B, determined according to results of testing models in a wind tunnel:

$$P_{wm} = B\pi R^2 \frac{\rho v_{n.reg}^2}{2}. \qquad (222)$$

For checking the strength of supporting device bearings under the effect of $P_{b\ max}$, we will consider wind velocity during a gust and the corresponding change in coefficient B. Then,

$$P_{wm\ max} = B_{gust}\pi R^2 \frac{\rho v^2_{gust\ max}}{2}. \qquad (223)$$

We will locate the moment of gyroscopic forces for maximum values of windmill rotation frequency and angular velocity of head reorientation. /227

Fig. 131 shows typical diagrams of supporting devices. In small electrical charging machines, sliding guide bushings and supporting rings of materials having a low coefficient of friction are used (Fig. 131, a). In larger motors, the head is connected to a supporting pipe installed in two radial-supporting bearings, located in the ipe of the tower (Fig. 131, b). With relatively high axial loads from the weight of the windmill and head mechanisms, a supporting bearing is additionally installed directly above the radial one (Fig. 131, c) or an adjustable caster is used (Fig. 131, d).

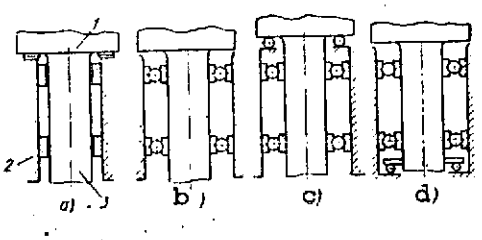


Fig. 131. Typical diagrams of supporting device: a. with sliding bearings; b. with radial-support bearings; c and d. with upper and lower support bearings; 1. head; 2. tower; 3. support pipe.

During recent years, a combined radial-support bearing with one or two rows of balls are used in a number of machines having a power of from 2 kW. This simplifies the design, reduces metal consumption, and eases placement of the parts of the starting and stopping mechanism. The advantage of these radial-supporting devices is the fact that they absorb any types of loads: axial and radial forces, and tipping moments.

A supporting device made according to the diagram shown in Fig. 131, a is applicable to the D-4 electrical wind machine (see Fig. 73). For decreasing the moment of forces of friction, the guide bearings are manufactured of bronze, and the support bearing is made up of several bronze and textolite rings. The head of the VBL-3 machine rotates in two radial-support bearings, in which the upper absorbs both radial and axial forces, and the lower absorbs only radial ones (see Fig. 45).

The supporting device of the D-12 high-speed wind motor is made according to the diagram with bottom location of the support bearing (Fig. 132). In the supporting device of the AVES-0.1 wind electrical machine, the upper support bearing absorbs all axial loads, while the guide bushing 4 (see Fig. 71) center the vertical tube of the head and absorb only radial forces. /228

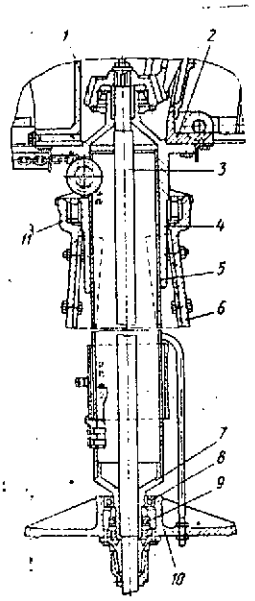


Fig. 132. Supporting device of a D-12 wind motor: 1. gear; 2. head body; 3. shaft; 4. support tail; 5. support pipe; 6. upper tower section; 7. pin; 8. radial bearing; 9. support bearing; 10. support plate; 11. roller bearing.

Combined support-radial rotating devices used in unified machines are most interesting. Fig. 62 shows the design of a unified device, whose outer ring is colocated with the worm wheel of a wind rose orientation mechanism. The inner ring is made to be separable so that, using inserts 9, it is possible to adjust the clearance between the rolling tracks of hot wire 10 and balls 3. As the wire wears, the amount of clearance is reestablished by decreasing the number of inserts. A diagram of this device is shown in Fig. 133, a.

For high power (20 kW and higher) wind-energy machines, supporting-rotating devices are better made two-rowed, with which the rolling tracks are not formed of wire rings, but are formed directly on the outer and inner rings (Fig. 133, b). A modified design for a device using two ball bearings (Fig. 133, c) is more convenient in assembly and adjustment.

We will examine the layout for calculating a single-row support-rotating device. The major task of the calculation is determining the maximum loads at the points of contact of the rolling bodies with the rings and determining the maximum contact stresses at these points, i.e. evaluating the load-bearing capability of the design [40]. Due to low circular velocities of rotation of the head, longevity calculations should be conducted only for high powered machines.

The summary load on the supporting device is

$$P_{apt\ max} = P_{ax\ max} + P_{rad\ max} + P_{m\ max}, \qquad (224)$$

where the components are determined by the action of the corresponding axial P_{ax} and radial P_{rad} forces and moment-creating forces P_m.

Taking the formulae using during calculation of normal bearings, we obtain /229

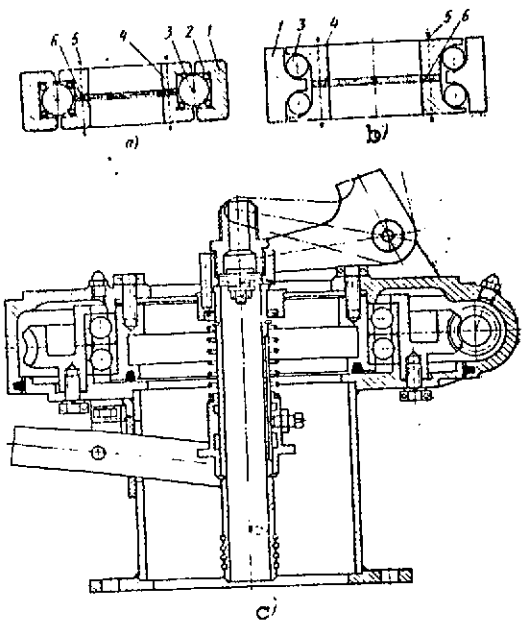


Fig. 133. Design of support-rotating devices: a. single-row; b. double-row; c. with standard bearings; 1. outer ring; 2. wire ring; 3. ball; 4. inner ring; 5. bolt; 6. inserts.

$$P_{ax\ max} = \frac{G_{hd}}{nm \sin \alpha_{ba}}; \quad (225)$$

$$P_{rad\ max} = \frac{(P_{b\ max} + P_{pl})k}{nm \sin \alpha_{ba}} \quad (226)$$

$$P_{m\ max} = \frac{k(M_g + M_{imb} + P_b h_{cp})}{d_{spt} nm \sin \alpha_{ba}} \quad (227)$$

where n is the number of balls; m is the number of inserts in simultaneous contact; α_{ba} is the angle of contact of balls with rings (usually, $\alpha_{ba} = 45°$); k is the coefficient of adjustment of radial load to axial; d_{ax} is the average diameter of a circle passing through the centers of the balls, in cm; P_{pl} is the force of pressure on the lateral plane.

The value of these forces for the most difficult working rates is determined from an aerodynamic and force computation of the machine. Then the load on the supporting device at points of contact of the rolling bodies with the tracks is determined according to an equation, which, after transformation, takes on the following appearance: /230

$$P_{\Sigma\ max} = \frac{C_y S \frac{\rho}{2} (\omega^2_{wm\ max} r^2_m + v^2_{max}) i_b k + (G_{wm} + G_{hd} + G_{or})}{mn \sin \alpha_{ba}} + \frac{k(G_{wm} l_{wm} - G_{or} l_{or} + J_{wm} \omega_{wm\ max} \omega_{2\ max})}{d_{ax} mn \sin \alpha_{ba}}, \quad (228)$$

where v_{max} is the maximum calculated wind velocity in m/sec.

Stresses in the material of the balls and wire are determined according to the known methodology (according to Hertz), used in calculation of bearings. Calculated contact stresses can be taken within limits of 15-25 n/mm^2.

The allowable static load per single ball is

$$P_{ba\ alw} = \left(\frac{\mu\nu P}{187}\right)^3 \frac{1}{(\sum\rho)^2}, \quad (229)$$

where μ, ν are the values determined according to table [40];

$$\sum\rho = \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22}$$

(ρ_{ik} is the radius of curvature of contacting bodies in mutually perpendicular planes).

In the majority of cases, the load proves to be within limits of 60-80% of allowable (lower values are for higher powered and high-speed machines). The clearance between the balls and rolling tracks with a device diameter of 0.3-0.5 m is taken at 0.1-0.3 mm. The hardness of the rolling racks, which are usually manufactured of carbon steels, after annealing is HRC 40-50; the balls, of type ShKh 15 steel, have a hardness of HRC 55-60. Extended operation of machines with these types of support-rotating devices showed that they are convenient in assembly and operation, easily adjusted and wear out very slowly.

The calculation of bearings used in support devices is not very difficult, since the velocity of rotation is very low (up to 1.2 rad/sec). Therefore, bearings are selected according to static load capacity.

§ 6. Calculation and Design of a Mechanical Transmission

The singularities of the work of the machine, caused by wind gusts and changes in flow energy and windmill rotation frequency, are considered during planning of any type of drive. The mechanical drive is usually used in machines with piston and screw-type pumps with H_{st} = 25-30 m and in some universal installations with a power up to 10 kW, in which the equipment (generators and working machines) is located in the lower part of the tower. For elements of a mechanical drive, it is characteristic to have long and thin shafts, relatively high initial moments of resistance, devices for damping overloading moments and, in a number of cases, significant flywheel masses of driven links, whose moments of inertia are commensurate with the moment of inertia of the driving link, i.e. the windmill, or greater than it. The latter occurs when using inertia accumulators. /231

Calculations for static strength of transmission elements, as for the entire wind machine, can be solved by the usual methods used in machine building [21, 33, 35].

Oscillogram investigations have shown that the least loaded elements of a transmission are usually the main shaft and high-speed gears, and the maximum stresses are characteristic for the vertical shafts.

Using the main shaft loading diagram shown in Fig. 134, we locate the summary stresses in the dangerous section I-I.

$$\sigma_{driv} = \frac{k_\sigma}{\varepsilon_\sigma}\sqrt{\sigma^2 + 4\tau^3} =$$

$$= \frac{k_\sigma}{\varepsilon_\sigma}\sqrt{\left(\frac{M_{g\ max} + G_{wm}l_{wm}}{W_{fl}}\right)^2 + \left(\frac{M_{obl}}{W_{fl}}\right)^2 + 4\left(\frac{M_{mo}k}{W_{tw}}\right)^2} \pm$$

$$\pm \frac{P_{frt}}{F_w} \leq \sigma_{-1}, \qquad (230)$$

where k_τ is the coefficient of overload because of torque moment.

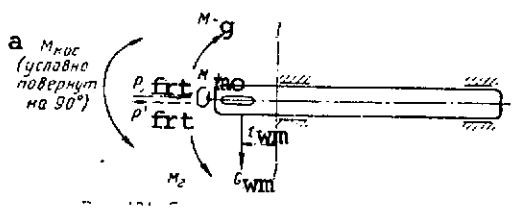


Fig. 134. Loading diagram of main shaft.

Key: a. M_{obl} (arbitrarily turned by 90%)

The main shaft of the machine having a windmill with a diameter of D_{wm} = 4 m and a speed Z_n = 6, over a year's operation undergoes approximately 10^8 loading cycles. When calculating shaft bearings, it is necessary also to consider the action of force P_{frt}.

Along with the aperiodic changes in the rotational moment, caused by changes in velocity and particularly the direction of the wind, constant pulsations of moment M_{mo} take place, caused by errors in manufacturing of the gears, imbalance of the windmill and others, and also reflecting singularities of the load [50]. The frequency of pulsation of the moment is shorter than or equal to the number of revolutions or the number of blades. In drive elements of the D-12 motor, the absolute value of the moment pulsation comprises 305-38% of the mathematical expectation of the moment. Upon engagement of the load to a direct current generator, the momentary value of moment M_{mo} exceeds its nominal value by 2-2.2 times (Fig. 135).

In machines with mechanical drive, torsional oscillations of the shafts arise at some working rates, but the coefficients

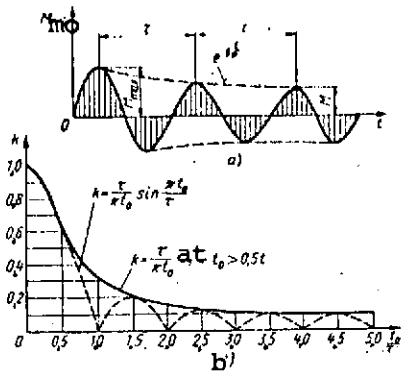


Fig. 135. Change in moment during torsional oscillations of the vertical shaft:
a. damping oscillations;
b. coefficient of overload.

of overload rarely exceed two. During gradual dampening of the oscillations, superposition of oscillations from two sequential impulses can take place. In high-speed machines with synchronous generators, during engagement of short-circuited asynchronous motors, the moment transmitted to the main shaft and gears can briefly increase by 2.3-2.5 times, and with short-circuiting, the momentary value of the coefficient K_{per} reaches 6.2. This must be taken into account during dynamic calculations. With a mechanical drive of sucker-rod pumps, performing a back-and-forth motion, unevenness of the acting forces is as a rule the maximum.

Maximum forces in elements of the transmission are frequently determined by overloads of the windmill. If the parameters, in particular the moment of inertia of the moving parts of the water lift, are known, it is possible to calculate overloads in the reduction gears, shafts and other elements, using known equations of dynamics and Carnot's theorem on equality of reserves in the kinetic energy of a system.

The moment transmitted during a wind gust from the windmill to the transmission will be /233

$$M_x = (M'_{mo.n} + \varepsilon J_{e.x}) i_x \eta_x = M'_{mo.n} + M_{mo.ex} \frac{J_{e.x}}{J_{e\Sigma}} i_x \eta_x, \quad (231)$$

where $M'_{mo.n}$ is the nominal moment adjusted to the shaft of the windmill; $M_{mo.ex}$ is the excess rotational moment; $\varepsilon = M_{mo.ex}/J_{e.\Sigma}$ is the angular speed of the windmill; $J_{e.x}$ is the summary moment of inertia of the masses from x to n, located in the section of the transmission examined, adjusted (equivalent) to the shaft of the windmill; $J_{e.\Sigma}$ is the summary moment of inertia of all masses of the mechanism, adjusted to the shaft of the windmill; i_x, η_x are the transmission ratios and coefficient of efficiency of the drives from the windmill to link x.

Knowing the rotational moments of the windmill during a gust $M_{mo.gust}$, and determining speed from the expression

$$M_{mo.gust} - M_r = J_{wm} \frac{d\omega}{dt} = J_{wm}\varepsilon, \quad (232)$$

it is possible to find overloads.

For a machine with a single screw pump, the maximum moment in the transmission is located according to formula (105).

The length of the vertical shaft is determined by the height of the tower. In transfer rates, caused by a wind gust or engagement of the load, torsional oscillations of the shaft can arise when the potential energy of the forces of flexion transfers into kinetic energy of rotation of its elements. If the shaft supports are located at a great distance from each other, the torsional oscillations are accompanied by cross-sectional oscillations. Therefore, the length of the shaft between supports must be maintained at no greater than 2.5 m.

We will examine forces acting in sections of the shaft, to the ends of which are applied loading moment M_r. In section $X = 0$, a twisting moment equal to M_r acts on the shaft; in section $(x + dx)$, the moment is $M_r + \frac{\partial M_r}{\partial x} dx$. Besides this, $X(x,t)dx$, causing oscillation of the shaft and, in accordance with the d'Lambert principle, the moment of inertial forces $R(x,t)dx$ also act.

$$M_r + \frac{\partial M_r}{\partial x} dx - M_r + X\,dx - R\,dx = 0 \tag{233}$$

or

$$\frac{\partial M_r}{dx} + X - R = 0. \tag{234}$$

Considering that $M_r = G_{sh}J_s \frac{\partial \phi_s}{\partial x}$, and $P_x(x,t) = J'_s \frac{\partial^2 \phi_s}{dt^2}$, we obtain the differential equation for torsional oscillations of the shaft

$$G_{sh}J_s \frac{\partial^2 \phi_s}{\partial x^2} - J'_s \frac{\partial^2 \phi_s}{dt^2} + M_r = 0, \tag{235}$$

where G_{sh} is the module of shear; J_s is the polar moment of inertia of the cross section of the shaft; ϕ_s is the angle of twisting of the shaft; $J'_s = \rho J_s$ is the polar moment of inertia of the mass of the shaft applied to a unit of its length.

The work of the motor in a machine with a piston pump causes the appearance of dynamic loads which are especially significant in a crankshaft-connecting rod mechanism and the rod of the pump

drive. Insufficient knowlege of these loads and imprecision of the methods of calculation used have had an effect on the reliability of the machines. Usually, during the use of such machines on shallow water sources, the balanced equilibrium of the pump and rod is not used. Therefore, during the piston's travel upward, the moment of the windmill is expended on overcoming the forces of friction in the drive and the pump, the weight of the rod and the column of water in the space above the piston, and in the pressure pipeline.

To simplify the task, we assume that the moment of the forces of friction in the tranmissions depends little on the position of the piston. Then loads on the elements of the motor will be determined primarily by forces transmitted from the pump. If the rods are tubular then, considering emersion in water, their weight is

$$G_{rd} = \pi \frac{l_{rd}}{4} [d^2_{o.rd}(\gamma_{rd} - 1) - d^2_{i.rd}\gamma_{rd}], \qquad (236)$$

where l_{rd}, $d_{o.rd}$, $d_{i.rd}$ are the length, outer and inner diameters of the rods; γ_{rd} is the specific weight of the material of the rods.

Taking the force of friction $P_{f.c}$ of the cuff on the wall of the cylinder during the piston's stroke upward as equal to 10% of the weight of the water column G_w and as equal to 5% during the stroke downward, we obtain the static forces acting on the upper part of the rod during its travel upward:

$$P_{st} = G_{rd} + G_w + P_{f.c} = G_{rd} + 1.1G_w. \qquad (237)$$

We will determine the inertial forces P_{in} arising during accelerations of the piston out of the following considerations:

Velocity v_p and acceleration j_p of the piston are

$$v_p = \omega_c r_c \left(\sin\varphi_c \pm \frac{\lambda}{2}\sin 2\varphi_c\right); \qquad (238)$$

$$j_p = \omega_c^2 r_c (\cos\varphi_c \pm \lambda\cos^2\varphi_c), \qquad (239)$$

where ϕ_c is the angle of rotation of the crank; $\lambda = r_c/l_{cr}$ is the relationship of the crank radius to the length of the connecting rod; ω_c is the angular velocity of the crank in rad/sec. /235

The maximal loads during the piston's travel upward are

$$P_{\Sigma\max} = P_{in^{\max}} + P_{st} = \frac{G_{rd} + G_w}{g} \times \left(\frac{\pi n_c}{30}\right)^2 r_c(1+\lambda) + G_{rd} + 1.1G_w \tag{240}$$

during the piston's travel downward, they are

$$P_\Sigma = \frac{G_{rd}}{g} r_\kappa \left(\frac{\pi n_c}{30}\right)^2 (1-\lambda) + P_{f.c} - G_{rd} \tag{241}$$

The coefficient of unevenness of loads acting in the drive elements over one double stroke of the piston

$$K_u = \frac{P_{\Sigma\max}}{P_\Sigma} = \frac{\frac{G_{rd}+G_w}{g}\left(\frac{\pi n_c}{30}\right)^2 r_c(1+\lambda) + G_{rd} + 1.1G_{wm}}{\frac{G_{rd}}{g} r_c \left(\frac{\pi n_c}{30}\right)^2 (1-\lambda) - G_{rd} + P_{f.c}} \tag{242}$$

must be equal to 10-25. If acceleration of the windmill during a gust is considered, the loads increase still further, limiting the strength and reliability of the drive elements.

High initial loads are also characteristic for machines with mechanically driven vibration type pumps. They are caused by the principle for lifting water used and by singularities in the operation of the motor. High stresses in the material of the vibration pipes have sometimes led to their destruction.

The following diagram of forces acting in the sections of the pipe can be set forth: tension forces from the inherent weight of the pipe and the column of water located in it; flexion and tension during cross-sectional oscillations; and flexion-compression from inertial forces arising as a result of a change according to the sinusoid law of velocity of reciprocal motion.

We will examine the effect of inertial loads caused by the transfer processes during changes in the velocity of the wind. In the type VB-3 machine, the effect of such loads is aggravated due to the relatively low moment of inertia of the windmill and the moving parts connected with it.

We will find the possible accelerations of the system for a case in which with v_c = 6 m/sec, the blades are rotated to the calculated angle, and the momentary wind gust comprises $1.6v_c$ according to characteristic $\bar{M}_{mo}(Z)$. We determine forces in the water lifting pipe out of two possible layouts of interaction which participate in the motion of the masses:

1) The column of water has little effect on dynamic overloads and has an influence due to friction along the tube wall; /236

2) Water moves together with the pipe, comprising sort of a unit mass (system). The initial data are $J_{wm} \approx 4$ kg·m^2; $\omega_{wm.n} = 40.3$ rad/sec; and the weight of the pipe with the water at H = 17 m is equal to 70 kg.

With the established rotation, the maximum accelerations of the pipe, corresponding to TDC and BDC, $j_{max} = r_c\omega_c^2 = 37.2$ m/sec^2. Inertial loads $P_{in} = (G_\Sigma/g)j_c$ (in addition to weight loads) in the upper section of the pipe with consideration for the mass of water comprises nearly 2.65 kn, and without considering the effect of the water column, it is 1.42 kn. The inertial forces in the established working rate exceed static loads from the effect of the weight of the pipe with the water by almost four times. The number of loading cycles with an amplitude σ_{max} corresponding to $P_{in\ max}$ in the calculated rate is equal to 385 per minute.

We will examine the action of acceleration of the tube during a wind gust. The moments of inertia of the moving parts, adjusted to the shaft of the crank are equal to: for the first layout -- $J'_\Sigma = 4.12$ kg·m^2, and for the second -- 4.15 kg·m^2.

$$\frac{d\omega_{gust}}{dt} = \frac{M_{mo} - M_r}{J'_\Sigma}. \qquad (243)$$

With $v_c = 6$ m/sec, $Z_c = 9.4$, and during a gust ($v_{gust} = 9.6$ m/sec), we get $Z_{gust} = (\omega_{wm}R)/v_{gust} = 5.9$. The increase in rotational moment is

$$\Delta M_{mo} = \frac{\pi R^3 \rho}{2}\left(\overline{M}_{Z_{gust}} v^2_{gust} - \overline{M}_{Z_c} v^2_c\right). \qquad (244)$$

For our case $\Delta M_{mo} = 12.15$ n·m. The total rotational moment M_{mo} is approximately equal to 20 n·m, and additional accelerations during a gust comprise 4.8 rad/sec^2. Therefore, during a gust, even in the case of the most favorable conditions, tangential accelerations will be relatively high.

The starting moment in the transmission when working with a belt water lift is equal to

$$M_{start} = \frac{Gr_{dv}}{ij} + \sum_{k=1}^{n} \frac{J_{wm}\varepsilon}{i_c}, \qquad (245)$$

where r_{dv} is the radius of the drive pulley; $G = G_u - G_d$ is the difference between the weights of the upward moving and downward moving sides of the belt.

When combining with a compressor, the dynamic overloads are not great, since the moment of inertia J_c of the moving parts of a compressor are incomparably small by comparison with the moment of inertia of the windmill.

For considering changing working rates of the transmission, we will use adjusted loads, equivalent to the acting changing load. /237 We will consider the changeability of rates with the coefficient k_e, which is selected on the basis of summarization of fatigue damages:

$$k_e = \sqrt[m_1]{\sum \frac{t_i}{T} \cdot \frac{\omega_i}{\omega_1} \left(\frac{M_i}{M_1}\right)^{m_1}}. \qquad (246)$$

The repetition of various rates is determined by time t_i, which we take according to a distribution curve of wind velocities. Definite values of ω_i and M_i correspond to each velocity.

Considering overloads and taking $m_1 = 3$, we obtain

$$k'_e = \sqrt[3]{\sum \frac{t_i}{T} \cdot \frac{\omega_i}{\omega_1} \left(\frac{M_i}{M_1}\right)^3 \left[1 + \frac{t_{ov}}{t_i}\left(\frac{M_{ov}}{M_i}\right)^3\right]}. \qquad (247)$$

In zones with velocities of $\bar{v}_a$ = 4-6 m/sec, we have k'_e = 0.78-0.85 respectively.

§ 7. Calculation and Design of Wind Motor Towers

The head with the windmill is mounted on a free-standing tower or on a mast with guy wires. If there are not special requirements set forth by the consumer or necessitated by the relief of the terrain, the height of the tower is selected from conditions such that height h from the surface of the earth to the tip of a lowered blade is not less than 3 m, and in the presence of obstacles nearby, no less than height h_{ob} of the latter. The minimum height of the support $h_{sp\ min}$ is determined using the expression

$$h_{sp\ min} = \frac{D_{wm}}{2} + h_{ob} - h_{c.hd} = \frac{D_{wm}}{2} + (3 - 4)\ \text{m}, \qquad (248)$$

where $h_{c.hd}$ is the distance between the upper part of the tower and the axis of the windmill.

Although for increased output and improvement of working conditions it is profitable to locate the windmill at a great height, this will frequently lead to an increase in specific expenditures. Therefore, the height of the support is determined with consideration of a combination of influencing factors by conducting variational optimization calculations (Chapter 4). During the stage of design research, height h_{sp} is usually set out of requirements for engineering safety and convenience in servicing.

Welded free-standing towers (Fig. 136), having an open grid construction and a square or triangular section in plan are more convenient than guyed supports during assembly and operation, more stable and have a constant frequency of inherent oscillations. Supports with guy wires have a lower weight, but with changes in the tension of the cables, the frequency of oscillations of the system changes and vibration can occur. These supports are less convenient during erection and repair. Three-sided towers of pipe and reinforced concrete supports are also used abroad.

Grids of triangular shape with braces possess the highest load-bearing capacity by comparison with all other grid systems. It should be emphasized that, regardless of the rigidity of bolted connections, the load-bearing capacity of struts in these supports is lower than struts of welded supports. /238

During the manufacture of free-standing supports of reinforced concrete, cylindrical towers with prestressed longitudinal reinforcements provide the best results. In these towers, the flexion moment has a smaller effect on strength.

The basic loads on a tower, with the exception of weight, are dynamic ones. These are loads from the effect of the wind on the tower, the windmill and the lateral plane. With consideration of the fact that the height of the tower is usually low, wind loads can be determined through normative specific loading q_n: /239

$$q_l = q_0 C_{aer} = \frac{v_c^2}{16} C_{aer}, \qquad (249)$$

and maximal loads can be determined with consideration for the coefficient of gustiness,

$$q_{l.max} = k_{ovl} C_{aer} \frac{v_{max}^2}{16}, \qquad (250)$$

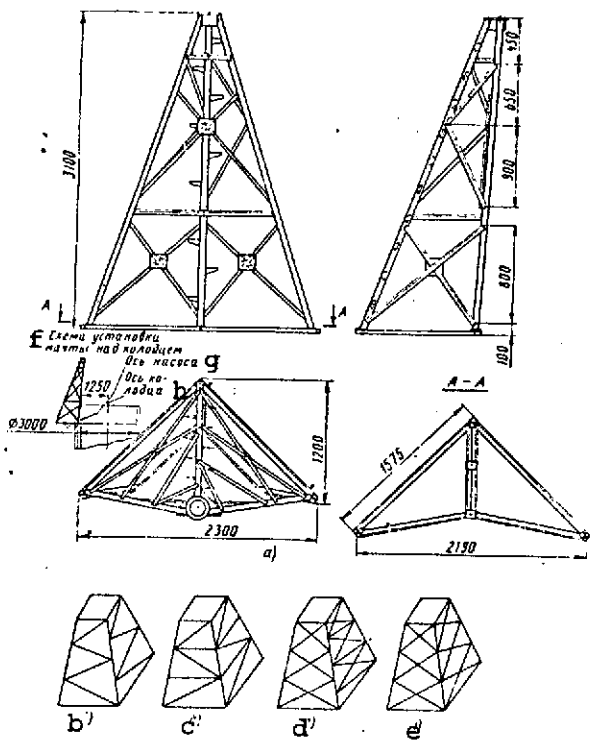


Fig. 136. Basic types of welded free-standing towers: a. design of leaning tower; b. herringbone grid; c. triangular grid with with struts; d. crossed grid with displaced joints; e. cross grid with combined joints; f. diagram of mast installation over a well; g. axis of pump; h. axis of well.

where C_{aer} is the aerodynamic coefficient; k_{ovl} is the coefficient of overload.

For regions of the most use of machines, it has been ordered to take q_0 = 450-550 n/$/m^2$, which corresponds to $v_{max} = v_5$, i.e. the maximum velocity repeated in a given area no less than once every 5 years. However, it must be kept in mind that the average velocity of the wind is recorded at meteorological stations over a 2-minute interval. In practice, velocities over several seconds are frequently needed, and they can be essentially higher. Evaluating the probability $P(v)$ of the fact that the wind velocity can be higher than the velocity found according to formula (4), the calculated velocity with the set norm of guarantee is determined.

Loads on the tower are connected with the effect of the wind flow and other disturbing forces on the windmill, the orientation system, and in some cases also on elements of the drive. Therefore, their combined interaction, considering not only the increase in summary loads, but also the possibility for the appearance of resonance phenomena must be examined. Resonance phenomena were observed in the VE-5 machine. The extremely sturdy and stable tower of the VE-5 machine was deformed as a result of resonance oscillations arising from rotation of the blades during the regulation process. Resonance is also dangerous in a case where the frequency of the inherent oscillations of the tower with the motor coincides with the rotation frequency of the windmill or with the number of impulses from the blade passing near the tower.

One of the singularities of a wind motor tower consists of the fact that its upper part (usually 25-35% of the overall height of the tower) is located in the shadow of the surface swept by the windmill, or in the zone of pressure (if the windmill

is behind the tower). Therefore, calculations of the tower according to summary effect of the wind without consideration for the effect of the windmill provide higher values of stresses and tipping moments by comparison with actual ones.

We will examine two calculated cases.

1. With a velocity of $v_{work\ max}$, the motor works. The summary loads on the tower in this case are equal to

$$P_{t\Sigma} = (P_t - P_t') + P_{wm}, \tag{251}$$

where P_t is the load on the entire tower without considering the effect of the windmill; P_t' is the reduction in load on the upper part of the tower due to the effect of the windmill. /240

The tipping moment is written thusly:

$$M_{tip} = k_{per}\left[C_{aer}\frac{v_{max}^2}{16}l_{aer} - C'_{aer}\frac{v_{max}^2}{16}(1-k_{aer}^2)\times \times (h_{sp} - R + l_{aer})\right] + B\pi R^2\frac{v_{max}^2}{16}h_{sp}, \tag{252}$$

where L_{aer} is the distance from the surface of the earth to the center of application of the resulting aerodynamic forces acting on the tower; C'_{aer}, l_{aer} are the position of the center of application of the resultant aerodynamic forces of the shadowed part of the tower and the coefficient respectively; k_{aer} is the coefficient considering the loss in velocity behind the windmill.

2. With a velocity v_{max} the motor does not work. In this case, the tipping moment M'_{tip} is equal to

$$M_{tip} = k_{ovl}\frac{v_{max}^2}{16}(C_{aer}L_{aer} + B_0\pi R^2 h_{sp}), \tag{253}$$

where B_0 is the coefficient of frontal resistance of a stationary windmill.

Forces in the rods of free-standing towers are approximately proportional to coefficient K_{gust}. Therefore, the dynamic component of the force can exceed the maximum calculated forces from the static effect of the flow by 60-80%.

If a model of a tower is tested in a wind tunnel and coefficient R_x of frontal pressure is obtained, the wind load on the tower is written according to the expression

$$P_t = R_x v_{max}^2 \left(\frac{h_{sp}}{h_{mod}}\right)^2. \quad (254)$$

Velocity v_{max} is taken as equal to 40 m/sec (for areas with $\bar{v}_a$ = 4-6 m/sec) and 50 m/sec (for zones with $\bar{v}_a$ > 6 m/sec).

The values of R_x for some types of towers are given in Table 28.

TABLE 28. VALUES OF R_x and $\bar{l}_{c.p} = \frac{x}{h_{sp}}$ FOR VARIOUS TOWERS AND MASTS

Type of tower	R_x	$\bar{l}_{c.p}$
Cylindrical with guy wires	0.0001	0.5
Four-sided grid pyramid	0.00085	0.43
Three-sided grid pyramid	0.0007	0.42
Three-sided solid pyramid	0.0017-0.0029	0.32-0.28

Frontal pressure on a windmill is determined according to formula (169) for the most difficult working rates (sudden gusts of wind, regulator did not succeed in kicking in), and the gyroscopic moment M_g creating additional flexion loads is determined according to formula (170). The moment from the unbalanced weight of mechanisms relative to the axis of the tower is determined using Fig. 130.

$$M_{flx.l} = G_{wm} l_{wm} - G_{sp} l_{sp}. \quad (255)$$

Static loads are created by the weight of the windmill, head and other mechanisms, and the inherent weight of the tower.

Frequently, calculation of a tower is not given the required /241
significance, only static loads are considered, and its elements are used out of construction considerations. The tower does have an essential effect on the reliability of the machine. A number of cases of destruction of a windmill, tail and tower from vibration have been recorded. This was also caused by incomplete consideration of a number of climatic factors: glaze ice, temperature affecting the forces in the guy wires, etc.

Calculation according to the actual load, which have not been sufficiently well known and were earlier considered with the

overall coefficient of reserve, allow excess strength reserves to be eliminated.

A tower is usually calculated according to the method of allowable stresses, which leads to incomplete utilization of the load-bearing capacities of elements of construction. We will examine the basic questions of calculation with a more progressive method of limiting conditions, signifying a move away from the overall coefficient of reserve and arbitrary calculated loads and possible approximation of loads to real expected maximum values. Calculation according to this method requires consideration of the dynamic action of the wind on the tower and the windmill, and also on elements of the automatic regulation and orientation systems.

We will set up two calculation cases:

1) According to the load-bearing capacity of the tower, when all elements must satisfy conditions of strength, stability or endurance;

2) According to the development of critical deformations in conditions of the worst combination of loads (bends, shifts or oscillations), limiting the possibility for further operation of the machine. Calculation formulae and sequence of calculation in this case differs little from those used during calculation of buildings and crane structures [21].

When calculating according to load-bearing capacity, we write the condition necessary so that the part does not fail thus:

$$P \leq F\sigma_{lim}m, \tag{256}$$

where P_c is the calculated force in the construction elements; /242
F is the geometric factor of the section of the element (area under tension or compression, moment of resistance under flexion and torsion); σ_{lim} is the limiting stress of the element; m is the coefficient of working condition, characterizing singularities in utilization of the tower.

The yield point σ_y is taken as the limiting stress of the material, corresponding to the moment of loss of load-bearing capacity of the tower or its element. The calculated resistance of the element is

$$R = k_u\sigma_{y.n}, \tag{257}$$

where k_u is the coefficient of uniformity of the material (for St. 3 steel, $k_u \approx 0.9$).

The possibility for increasing the actual operational load $N_{o.a}$ by comparison with the normative one N_n is evaluated with the coefficient of overload k_{ovl} (for low-speed machines, $k_{ovl} = 1.3$, and for high-speed ones, $k_{ovl} = 1.2$). Then

$$\frac{k_{ovl}N_n}{F} = N_{o.a} \tag{258}$$

or

$$\sigma_n k_{ovl} = \sigma_{y.n} k_u m;$$

$$\sigma_n \frac{k_{ovl}}{k_u m} \leq \sigma_{y.n}. \tag{259}$$

The amount $k_{ovl}/(k_u m)$ is the coefficient of reserve for stress from normative loads σ_n with respect to the normative yield point $\sigma_{y.n}$.

The second limiting condition for compressed elements is necessitated by the requirement that flexibility of the elements does not exceed that allowable, i.e.

$$\lambda c = \frac{l_c}{r_{min}} \leq \lambda_{alw}.$$

where l_c is the calculated length of the element; r_{min} is the minimum radius of inertia of the element's section; λ_c and λ_{alw} are the calculated and maximum allowable flexibility of the element.

Consideration of changing loads which are repeated many times is conducted according to the equation

$$\sigma = \frac{k_{ovl}P_n}{\gamma_\sigma F_{nt}} \leq R \tag{260}$$

$$\sigma = \frac{P_n}{\gamma_\sigma F_{nt}} \leq \sigma_{alw} = \frac{\sigma_{y.n}}{n}, \tag{261}$$

where P_n is the force in the element from the effect of nominal load; F_{nt} is the area (net) of the element's section; $\gamma_\sigma = \sigma_{\rho\beta}/\sigma_{y.n}$ is the coefficient representing the relationship of the fatigue limit (with given ρ and β) to the yield point; $n = 1.3\text{-}1.4$ is the coefficient of strength reserve. /243

The coefficient γ_σ is determined according to the expression

$$\gamma_\sigma = \frac{1}{(\alpha\beta \pm a) - (\alpha\beta \pm b)\rho},$$

where α, a and b are constant values (for towers, $\alpha = 1.0\text{-}1.1$; $a = 0.9$; $b = 0.3$ are used); β is the coefficient of working rate, equal to 1.15-1.25; $\rho = \sigma_{min}/\sigma_{max}$ is the coefficient of asymmetry of cycle.

The base (foundation of the tower is calculated for stability for uprooting of the foundation). The calculation is based on an examination of conditions of equilibrium of the maximum external loads uprooting the foundation and forces preventing uprooting. The equation of equilibrium is expressed in the following form:

$$(\gamma_{sl}V_{sl} + S_f f_0 + 0.9G_f)k_{cn} \geq P_{upr}, \qquad (262)$$

where γ_{sl}, V_{sl} are the weight of the soil in the volume of the obelisk formed during excavation of the soil; $S_f f_0$ is the force of adhesion of the soil along the shear surface of the obelisk; G_f is the weight of the foundation; k_{cn} is the coefficient of working condition of the foundation.

Specific pressures p on the soil beneath the base of the tower foundation are also determined:

$$p = \frac{P_\Sigma}{F_f} = \frac{G_{mo} + G_{sp} + G_f + P_{w.max}}{F_f} \leq p_{alw}, \qquad (263)$$

where F_f is the area of the foundation foot; $P_{w\ max}$ is the load on the tower leg under the action of the maximum wind velocity along a cross sectional diagonal.

The allowable specific pressures p_{alw} depend on characteristics and condition of the soil. They change from 15 n/cm^2 for weak water-saturated soils up to 60 n/cm^2 for solid sandy and clay soils.

§ 8. Use of Nonmetallic Materials

Plastics and reinforced concrete are ever more widely used as construction materials in wind machines for manufacture of blades, towers, bushings and bearings. Along with a reduction in metal consumption and labor consumption of manufacturing parts and improved operational indices, with the use of these materials, the attainment of simple methods for identical blades which are resistant to atmospheric phenomena and mechanical defects is made easier. In connection with the operating conditions of wind machines, which make organization of constant servicing more difficult, it is necessary to move away from frequent lubrication of supporting devices and use self-lubricating bearings. For pasture machines, use of reinforced concrete towers, which during the periods when the watering points are not functioning are not dismounted while the basic components and mechanisms of the wind machine are subjected to preserving, is frequently justified. /244

Blades of pressed fiberglass are used in domestic and foreign machines. The load-bearing elements of the blade are the fiberglass skins or box longerons. Plastic foam is usually used as the filler. The absence of metal spars and ribs simplifies manufacture of the blade. Abroad, fiberglass is reinforced with steel or aluminum wire, and is also fitted with a metal load-bearing longeron (spar).

Plastics reinforced with fiberglass on polyethyl resins possess high mechanical characteristics. The tensile and cross breaking strength of fiberglasses on a base of PN-1 resin reaches 0.2 kn/mm^2; they are resistant to the effect of repeated loads. The strength of fiberglass under low frequency cyclic loads with the asymmetrical cycle taking place in blades is close to the strength of aluminum alloys and reaches 0.13 kn/mm^2. Blades of fiberglass are not very sensitive to cuts or the action of abrasive particles. Considering, however, that the windmill is a responsible component, the lowest fatigue limits (up to 0.07 kn/mm^2) should be selected when calculating blades.

The use of fiberglasses decreases the weight of the windmill, increases reliability, simplifies servicing, since the blades are less damaged during work and transportation, and does not need painting. Pressed blades have higher quality surface and coefficient ξ by comparison with blades having a metal skin. Of the two methods for pressing blades: 1) under high pressures and temperatures using hot setting resins and 2) under normal temperatures using cold setting polyester resins (PN-1, PN-4), the most preferred is the second, in which time and expenses on manufacture are reduced. However, with the second method for pressing, the strength characteristics and resistance of the blade against the effect of moisture and temperature jumps are somewhat reduced (by 12-15%0.

Of the two typical blade designs (Fig. 137, a and b) of machines having a power of 1.5 kW, the first (Fig. 137, a) does not have a longeron, and its load-bearing skin is made of changing thickness, decreasing from 10 mm at the hub to 2 mm at the periphery; the inner hollow is filled with plastic foam. This blade design is preferable for relatively thin blades and is convenient in that one molding press is required for manufacture of the blade. Blades of constant width (Fig. 137, b) have a load-bearing longeron, which is also obtained by the cold pressing method. This allows the skin thickness to be decreased, but the technology of manufacturing the blade was made more complex. Blades of larger machines are fastened to the hub with universal tips, which are connected to the blade with bolts. /246

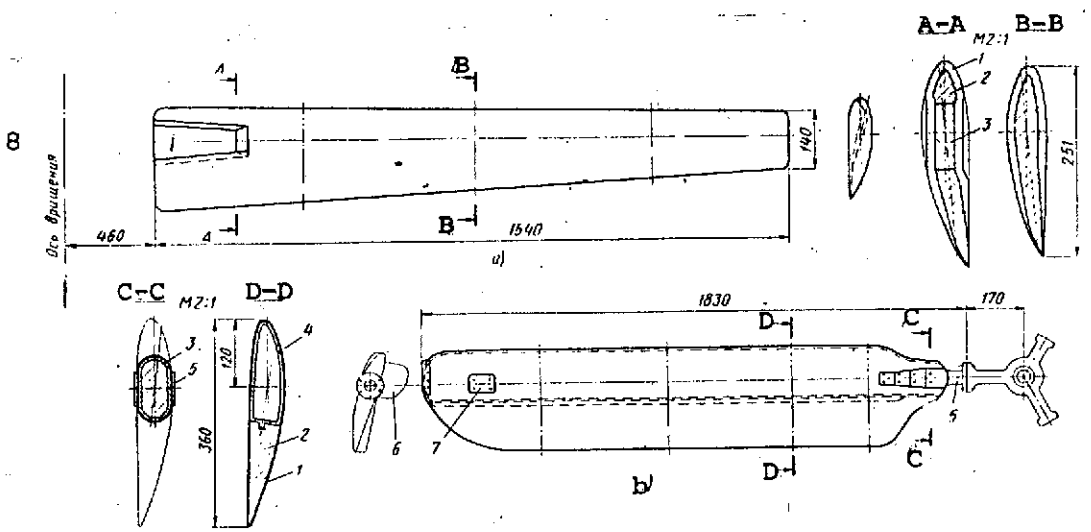


Fig. 137. Blades of fiberglass: a. without longeron; b. with longeron; 1. skin of glass fabric; 2. plastic foam filler; 3. textolite insert beneath connector; 4. longeron; 5. connector; 6. tab; 7. balancing weight; I. slot for fastening connector; 8. axis of rotation.

Under strict observation of the engineering process, the difference in the weight of pressed blades going into a set is relatively small. Thus, with D_{wm} = 4 m, it does not exceed 2%, and deviation in the value of the static moment is no greater than ±0.7 n·m. For hot pressing, this deviation is still smaller. Selective choice of blades during formation of sets allows their balancing to be avoided.

Testing of blades and accumulated experience allows their incorporation to be recommended. With an identical volume of

space, the cost of manufacturing blades of fiberglass is 15-20% lower than metal ones.

A wind machine has many bushings and sliding bearings, for which thermoplast polymers with a low specific weight and coefficient of friction, and which do not require lubrication, are relatively cheap and easily machined, are effectively used. Some of their deficiencies (low heat conductivity and heat resistance, high coefficient of linear expansion) are not essential for the working conditions in wind motors. The most long-lived of these polymers is fluoroplast-4: it is not wetted by water, and when filled with graphite, has a coefficient of friction of $f = 0.02$ (the coefficient of dry friction on steel is equal to 0.04-0.07). It is used for regulator bushings and in the supports of the water lift and drive. In more difficult conditions (at high temperature and dynamic forces) AG-1500 stannic graphite works best.

Replacing the metal support with reinforced concrete reduces expenditure of metal by four times and greater. Depending on the purpose and layout of utilization, the tower can be made tubular with one or two bracing guys and a hinged device in the lower part; free-standing with its section changing with height and a rigidly fastened foundation part or a frame type with reinforced concrete elements fastened together with bolts. The strength of supports made of prestressed reinforced concrete can be significantly increased by centrifuging.

Supports of the first type can be recommended for machines of medium power. The expenditure of metal on manufacturing these supports is reduced by 6-7 times by comparison with the expenditure of metal for manufacturing a metal support. Supports of the second type are expedient to use for small electrical and pumping machines. For conservation of the motor, these supports are not dismantled. Supports of the third type are universal, although /247
their assembly is complex. With an output of 3-4000 per year, the cost of this support is equal to the cost of a metal one or somewhat lower.

§ 9. Unification of Wind Machines

The many various regional conditions and consumers bring about the necessity for using machines which are different not only in power, but also in the type of pump, speed of the motor, height of the support, etc. Therefore, unification of wind machines is very real. The necessity for unification is also dictated by the relatively small series of machines of each type and size, grouped servicing of installations, especially in pastures, their dispersal, difficulties in creation of repair bases and supply of spare parts.

Machines produced earlier did not have unified parts. Therefore, it was difficult to use leading technology and decrease the cost of machines, and to increase reliability and the quality of manufacture. An increase in the spare parts list complicated operation.

The tasks of unification consist of creating universal parts-blocks (blades, tower, supporting devices, etc), which can be used in machines of a definite power range, regardless of their purpose; of development of parts (reduction gears, regulation mechanisms and others), possessing design and technological acceptability, for the purpose of using them in machines which differ in speed, drive type and so forth; and of reducing the list of parts. A reduction in listing is achieved not only be designing parts (drive elements, blade fastenings, electrical control panels, etc) as independent mechanisms adaptable for utilization in various machines, but also as elements which are built into more complex sets.

The solution of these problems does not remove the question of decreasing the number of types of machines and unification of parts for installations of various purposes (pumping, electrical, distilling, etc.).

Works in recent years, in the course of which VIESKh [Scientific Research Institute of Rural Electrification] and other organizations created a system of unified machines, showed great possibilities for the unification.

The degree of unification can be evaluated, as is usually done, with the coefficient of acceptability k_{ac}, i.e. the relationship between the number of borrowed parts S_B to their overall number S_{oa} in the machine (during determination of coefficient k_{ac}, parts of standard pumps and generators are not considered).

$$k_{ac} = \frac{S_B}{S_{oa}} 100\%. \qquad (264)$$

If there is a unified series of wind motors, the coefficient of unification k_u, equal to the relationship between the number of unified parts S_u to their overall number S_{oa} in the motor, is determined by comparing it with one of the motors of the series, taken as the base. /248

Not remaining on general principles, we will examine some actual examples of unification of machines which have gone into the machine system for complex mechanization of agricultural production. The coefficient k_u of certain motors comprises 0.7-0.75, which increases the level achieved by foreign wind

technicians. A characteristic example of parts unification is unification of the reduction gears in the "Buran" machine: the upper (see Fig. 57, a) and lower (see Fig. 57, b) reduction gears differ in the end part of the shaft and working position. For high-speed windmills with a diameter D_{wm} = 3-5 m, a regular connector (see Fig. 137), with which the blade is fastened to the spar is used, which simplifies assembly and eases replacement of damaged blades.

Intermediate type unification is achieved by using two base models: one for motors with w wind rose system of orientation, and the other for machines with an orientation system using a tail empennage. Replacing the compressor of a "Vikhr'" machine with a generator, and a pneumatic pump with a centrifugal one provided the possibility for creating the "Berkut" electrical machine, whose remaining parts remained without changing. A unified support, whose ring was colocated with the worm wheel was used in the orientation mechanism of this machine. The set velocities of ω_2 of head rotation and the required values for moments are obtained by changing only the diameter of the wind rose (through the length of the spars) or their speed (through the angles of blade setting).

Unification of towers is achieved by using normal sections: a lower one with a hinging device, an intermediate one of angle steel or pipe and an upper tubular section, whose length depends on the diameter of the windmill. Support flanges and butted section joints were unified.

§ 10. Automation of Wind Installations

The proper scope and methods of automation depend on the type and purpose of the installation, its power and conditions of use [45]. They are brought about by the basic requirement for the machine -- to work always when there is a wind and a consumer which can consume energy (water) at a given moment or after some, relatively brief interval in time.

An installation on a farm is usually under the observation of a mechanic and therefore the introduction of a set of additional automation devices is not always justified. The simplest mechanisms for starting a motor into operation upon a lowered water level in the tower or stopping it when the tower is filled might be an exception.

The power of the motor, up to engagement of the regulation /249
system into operation, proportional to the cube of the wind's velocity, and rotation frequency of the windmill is proportional to velocity. The characteristics of the majority of water lifts are such that the power of the motor is not fully realized at all wind velocities. It is possible to use devices which automatically correct the characteristic so that with increase in wind velocity

and windmill rotation frequency the required power is close to the power of the motor. Use of these devices is not, however, usually justified due to the complexity of the machine and the small effect which they yield.

Systems for orientation and limitation of rotation frequency and power are attributes of the minimum level of automation. Some machines ("Buran," "Chayka") have devices for automatically locking the motor in the stopped condition during storms, stopping or starting the motor depending on the level of water in the tank (VVU-3, VB-3), and relieving and controlling the operation of the pump ("Vihkr'" and others).

With installations working in pastures in the steppes, deserts and semidesert regions, where sources have low yield, one of the tasks of an automatic device is limitation of the quantity of water raised within limits necessary for consumption and reserves. The intensity of removing water from these sources must be limited so that the water level in the well does not become lower than the set level, or otherwise the source might be spoiled. Draining water into the source when the tank is overfilled in accordance with animal breeding requirements is not always possible.

In the majority of cases, the parameters determining kick-in of the automatic device might be the level of water in the tank or in the well. When using fill-in wells, where the fresh water is located on a saline layer, the fresh water must be pumped out as it arrives, without touching on the mineralized layer. If a pump with a laminar water collector is set so that its intake valve is placed above the lever of the break in layers, after the fresh water is pumped out, the pump begins to work at idle, but the motor does not stop. However, working without liquid is inadmissible for centrifugal, piston and screw-type pumps, since it causes high wear in cuffs, rings, seals and overheating of the electric motor windings. Therefore, it is better to use devices which turn off the pump or the motor when the level of water in the well is lowered or when saline water enters the pump. In machines with electrical drive, their construction is simpler than in machines with drives of other types.

If the installation is used for charging batteries or raising water from several wells, alternating connection to the generator of the more discharged storage batteries or cyclic engagement and disengagement of the pumps upon establishment of the water level in the well or, on the contrary, upon an extremely decreased level respectively is provided by automatic devices.

Because of the specifics of a wind installation, automatic /250
devices must be maximally simple and reliable, and no external power source must be required for their drive, i.e. the automatic device must work under the action of the force of gravity, the

weight of the water or its ejection force. In electrical machines or in machines with pneumatic or electric pumps, it is possible to use devices which are driven using compressed air or which are kicked in when the power fed by the generator is stopped.

In a mechanism for automatically starting and stopping the motor depending on the level of water in the tank (Fig. 138), the principle of communicating vessels is used. A drum, connected with the tank by a hose, is suspended on a lever and balanced with weight 6. Having become filled with water, the drum moves downward and stops the motor, acting on the rotating blades. When the water level falls, the drum is drained, and weight 6, returning lever 5 into its initial position, rotates the blades to their starting angles. This system is simple in installation, but difficult to use when the tank is removed from the motor by more than 2 m, as well as under winter conditions.

The advantage of weighted or floating devices (Fig. 139) is that the reservoir can remain at a somewhat greater distance from the motor, although they are also intended for work with the temperature of the surrounding air higher than zero. The mechanism with a float valve (Fig. 139, c) can be located inside the tank or outside it, and it is easier to heat. The capacity of the drum with this is determined by the force necessary for stopping the motor.

A similar device can be used by connecting its operation with the level of water in the well where a drain pipe with a valve which opens its hole upon a decline in the water level is additionally lowered. With this, the water is pumped along a closed main line: well → pump → pressure pipe → drain pipe → well, and in the presence of a reservoir: well → pump → pressure pipe → → reservoir → drain pipe → well.

In the layout of an automatic system for a pneumatic installation, consisting of sensors and valves installed in the reservoir or the source (Fig. 140) provides, if necessary (for instance, with limited tank volume), engagement of a compressed air accumulator into the system. The singularity of the system is that the machine is not forceably stopped, and it always works or is ready to work. The mutual position of the well, tank and motor can be any. On main line 8, along which air is supplied to the pump, are installed switches 5 and 10 with floats having two positions: I -- the receiver is connected with the distributor device 7 of the pump; II -- the receiver is connected to the accumulator. The receiver-accumulator main line is opened when the leverl of water in the well drops lower than that set (diagram in Fig. 140, a) or when the tank is filled (diagram in Fig. 140,b). Water flows into the tank along pipe 9. If pressure in the system exceeds that set, valve 2 releases excess air. /251

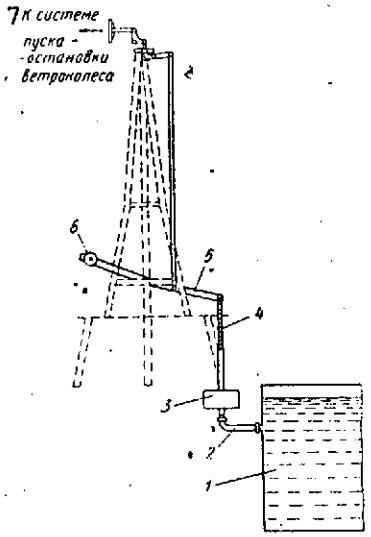


Fig. 138. Diagram of mechanism for automatically starting and stopping a wind motor, used in domestic pumping wind installations: 1. tank; 2. hose; 3. ballast drum; 4. pull rod; 5. lever; 6. weight; 7. to the windmill starting and stopping system.

The diagrams presented do not exhaust all the variations for switching and controlling devices, they are in each actual case determined by the tasks set. If the volume of the tank is sufficiently high, it is not expedient to install a compressed air regulator, although it increases the coefficient of utilization of the possible output of the machine.

When using a group of low-yield wells, the layout is also usable for automatic distribution of the air fed to the pumps, depending on the water level in the wells; in electrical machines, the air lines are replaced for this with electrical lines. If the /252
setting parameter is salt-containing water, a sensor which reacts to concentration of salt is installed on the suction main line of the pump. When saline water arrives, it stops the feed of power to the pump or releases air from the pump into the atmosphere.

Automation is more difficult for the medium and high power machines used on farms or in irrigation systems, as well as grouped machines working in parallel with a network or together with a non-wind station. The storage of electric power in large quantities is still not effective. A wind motor, however, works according to an uneven graph, and for many consumers power must be supplied at a definite time and in set quantities. Isolated wind-electric pumping machines must in a number of cases work together with non-wind stations, because of which large capacities for water reserves for the most extended periods of calm are not always economic.

The task of automating combination installations is that of providing rapid engagement of the non-wind station when the power of the wind motor is insufficient for carrying the entire load, or disengaging the reserve station if a stable increase in wind velocity is observed and the machine develops the required power. Automatic devices (regulators) for motors are adjusted so that with their parallel or combined operation, redistribution of moments is assured for the purpose of using all the power of the wind motor and abetting it with the required (according to loading /253
conditions) power of the internal combustion engine. Current

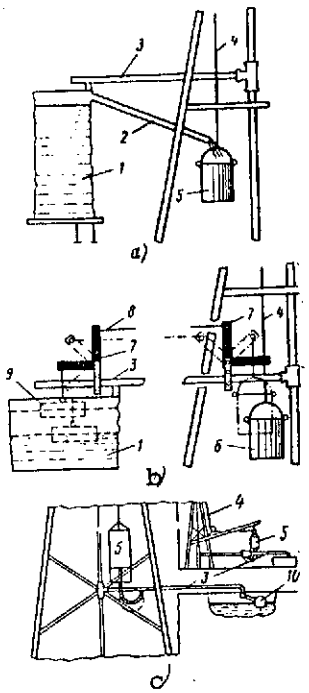


Fig. 139. Diagram for automatic devices controlling the operation of pumping wind installations: a. weight type device; b. float type mechanism; c. valve type device; 1. tank; 2. drain hose; 3. pressure pipe; 4. starting-stopping cable; 5. weighting vessel; 6. counterweight; 7. lever; 8. cable; 9. float; 10. float valve.

frequency and voltage of the generator are automatically adjusted.

To simplify servicing of installations and reduce operational expenditures, it is necessary to provide automatic emergency protection which assures stoppage of the motors with an extreme increase in generator rotation frequency, high water temperature and others. The installation of a remote control and signal panel in the apartment of the duty mechanic reduces the number of operational personnel.

In machines with a power of up to 25 kW, the use of a single generator driven by two primary motors simplifies the electrical part and automation system and reduces energy losses, increasing its output by 506%. The automation layout of the D-12 wind electric station, developed by VIESKh and the Diesel Institute (Fig. 141), provides maintenance of the normal current frequency with the windmill rotation frequency changing and provides the best utilization of the energy of the inertial accumulator due to the use of an automatically controlled stage-free variable-ratio transformer; redistribution of load between wind /254 and internal combustion motors; starting and stopping of the diesel depending on the power of the windmill, reflected by the reserve in kinetic energy of the accumular, and also upon feed-in of an impulse through the relay circuit to the electric starter or stopping device; protection of the diesel from overload, monitoring and maintenance of a set working rate for the diesel and its protection against destructive conditions connected with a drop in lubrication pressure or an increase in the temperature of cooling water of shaft rotation speed; and monitoring of the levels of fuel, oil and water with input of a signal for stopping the engine. The relay part of the layout is executed in the form of local and remoted automatic device panels using standard relays and apparatuses. It also provides the possibility for synchronizing two motors for parallel operation.

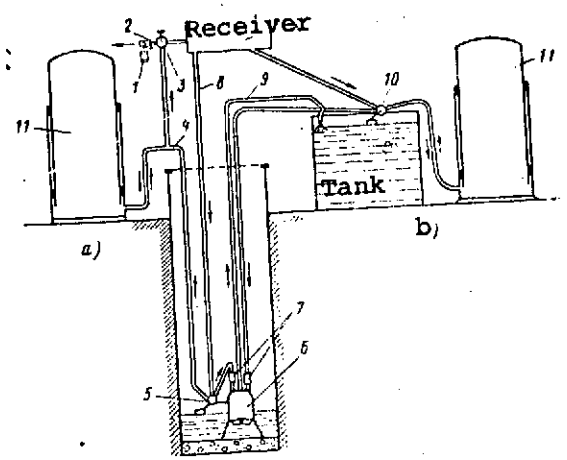


Fig. 140. Diagram of automatic system regulating operation of pneumatic pumping installation: a. with setting parameter being water level in source; b. same, according to level of tank fill; 1. safety valve; 2. valve; 3. switch; 4 and 8. air lines; 5 and 10. floating sensors; 6. pump; 7. distributor device; 9. pressure pipe; 11. compressed air accumulator;

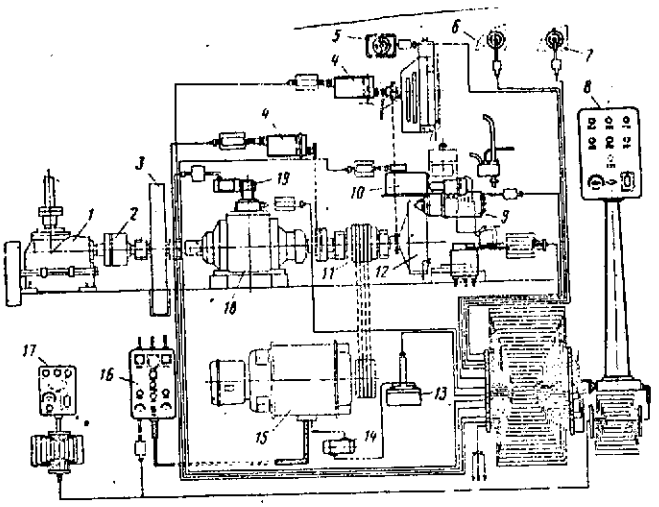


Fig. 141. Diagram of automation of the D-12 wind electric machine with a reserve diesel: 1. wind motor reduction gear; 2. freewheeling clutch; 3. inertial accumulator; 4. centrifugal relay; 5. water level relay; 6. oil level relay; 7. fuel level relay; 8. local control panel; 9. electric starter; 10. stopping device; 11. freewheeling clutch-pulley; 12. diesel; 13. storage battery; 14. rectifier; 15. generator; 16. distributor panel; remoted control panel; 18. infinitely variable drive; 19. servomotor.

When using wind machines with electric drive for irrigation and drainage, their parallel operation is most effective. The scope of automation in these machines must be as broad as possible, and both local and remote control must be provided. This is caused by the fact that the indices of a group of machines, including those in combination use with diesel engines or a power network, will be better with the wind machines located at a significant distance from each other, which provides more even power fed to the system by the machines, and better operating conditions for the pumps.

The introduction of automatic monitoring of parameters increases reliability of systems.

PART 5. ENERGY-PRODUCING AND TECHNICAL-ECONOMIC CALCULATIONS OF WIND INSTALLATIONS /255

Chapter 11. Energy-Producing and Balancing Calculations of Wind Installations. Selection of Storage Capacities

Wind power-producing machines work according to an uncontrollable graph, and therefore machines intended for mechanization of water supply, illumination of objects, and drive of some working machines must be combined with storage capacities, electrochemical storage batteries or reserve non-wind installations which compensate all or part of the power of the wind machine during periods of energy-producing calms and low wind velocities.

This requirement is not extended to machines which are used for draining over moist sections and electrical stations working in parallel with a powerful network or with thermal stations having a significantly higher power than the wind electric station. In these cases, all the possible output of the wind energy-producing machine, determined by its characteristics, regulator adjustment and wind rate is successfully used, and energy-producing calculations do not represent great difficulties, since the major task boils down to locating the summary usable output of energy over the selected time interval. For this, the number of hours of the machine's operation with wind velocity of v_0, $(v_0 + 1)$ and so forth are located according to wind velocity repetition curves. Repetition of velocities will lead beforehand to the height of location of the windmill's center from the earth's surface.

Knowing the operating rates of the machine at various wind velocities and the power developed, energy is computed.

$$A = \int_{t_{v_0}}^{t_{v_{work\ max}}} N_{wmv_i} \eta_{mechv_i} \eta_{elv_i} t_{v_i} \quad kW \cdot h. \qquad (265)$$

If the machine is used in the layout of parallel operations with a network, the rotation frequency of the windmill is practically constant and is determined by the frequency of the network. Therefore, in a wide range of wind velocities, the motor works not with the calculated number of moduluses and, consequently, with less power.

In practice, it is most frequently necessary to conduct calculations for determining the capacities of pumping machine reservoirs which are necessary for providing water to the consumer with a definite norm of guarantee P. With respect to methodology, they are identical to calculations for storage batteries. However, in this case it is necessary to additionally consider the /256

coefficient of efficiency of the battery, the allowable degree of its discharge and also limitation in the amount of charging current, which does not always provide the possibility for using all the power of the motor.

In the planning research stage, the capacity of reservoirs can be calculated according to the simplified formula

$$W = B_a(1 + t_{calm\ max})\ m^3, \tag{266}$$

where B_a is the average daily water consumption in m^3, $t_{calm\ max}$ is the maximum period of energy-producing calm with a given type of machine in days.

However, the result proves to be extremely approximate. Besides this, it is impossible to determine the optimal volume of a tank by economic and other considerations according to expression (266), since an attempt to create a water (energy) reserve for periods of the most extended calms, which are usually repeated no more frequently than twice per year, requires very large tanks or storage batteries and leads to an increase in capital expenditures.

The methods of calculation according to differential-integral curves, providing chronological comparison of output and consumption of power (water) by stage and computation of its surpluses or deficits should be considered the most accurate, although they are labor consuming [13, 17, 32, 39].

For a pumping machine, the equation for balancing the volumes of water pumped and consumed over the calculated period will be the following:

$$W_t = W_0 + \int_0^t Q\,dt - \int_0^t B_a\,dt, \tag{267}$$

where W_t is the volume of water contained in the tank at some moment in time t; W_0 is the volume of water contained in the tank at the initial moment in time t_0; $\int_0^t Q\,dt$ is the total water feed of the machine over time $t - t_0$; $\int_0^t B_a\,dt$ is the total water consumption over time $t - t_0$, and that for the balance of power in the general form for wind electric machines with a storage device is

$$A_t = A_{st} + \int_0^t A_{use}dt - \int_0^t A_{con}dt, \qquad (268)$$

where A_t is the quantity of power stored in the storage battery in kW·h; A_{st} is the quantity of power in the storage battery at the initial moment in time t_0; $\int_0^t A_{use}dt$ and $\int_0^t A_{con}dt$ are the useful output of power by the machine and its consumption over time $t - t_0$ in kW·h. /257

The difference between balance calculations for power and water volume is that in the first case we operate with usable power developed by the machine, and in the second with productivity, i.e. water fed by the pump.

We will examine problems which must be solved in the course of balance calculations for water volumes when using pumping wind installations. The major task of such calculations is determination of the minimal necessary volume of the tank for consumers, characterized by various graphs of water consumption; and establishment of the economically advisable, i.e. optimal norms of providing the object with mechanized water supply and the reservoirs corresponding to it. With this, in the majority of cases the major criterion of optimization is specific adjusted costs.

Along with solution of the main problem, optimal relationships between the water consumer, productivity of the machine and volume of the above-ground reservoir must be outlined, evaluation of losses during the most extended idle periods of the machine as a result of an absence of wind must be given; and optimal correlations between reservoir volume and fullness of the well must be found.

Therefore, the next stage after optimization of parameters of the machine is optimization of the storage device, which essentially affects the effectiveness of the entire installation.

We will examine the furthest step in calculations of storage capacity. We will determine the total output Q_Σ of the machine over period T thusly:

$$Q_\Sigma = T \int_{v_0}^{v_{work\ max}} f(v)F(v)dv, \qquad (269)$$

where $f(v)$ is the dependency of the productivity of the machine on wind velocity, pressure and other conditions determined by its design singularities; $F(v)$ is the chronological course of wind velocity in the region investigated.

If at moment in time t the quantity of water in the tank is equal to zero, and the feed of the machine is less than consumption, deficit D arises. If, on the contrary, $W_0 + (Q_\Sigma - B_s) > W$, water surplus S appears, and cannot be stored, due to the limited volume of the tank selected, /258

$$S = W_0 + (Q_\Sigma - B_s) - W, \tag{270}$$

where Q_Σ is the output of the machine over the interval in time between two water consumptions in m^3; B_s is the single consumption of water.

With $W_0 + (Q_\Sigma - B_s) < W$

$$S = 0.$$

The consumption of water usually takes place constantly and is a random process, whose characteristics can be determined using probability statistical methods. In pastoral conditions, the major consumption of water occurs during the hours of livestock watering and therefore for simplifying calculations it is possible to assume that the total or partial emptying of the tank (i.e. water consumption) coincides with the watering time and occurs two times per day ($B_s = B_d/2$).

If the total quantity of water required over the period of adjustment is equal to B_Σ, the degree of provision of a mechanized water supply by the wind installation can be determined according to the expression

$$P' = \frac{W_0 + (Q_\Sigma - S)}{B_\Sigma}. \tag{271}$$

Provisionability P shows how many adjustment periods out of the number of calculated ones give the indicated degree of water supply provision P.

The norm of provisionability is

$$P = \frac{m - 0,5}{n} 100\%, \tag{272}$$

where m is the order number of the sample member with the positions of the values of the members in increasing order; n is the total number of members in the sample.

Calculations made in relative units, i.e. in parts of the maximum possible daily output of the machine, taken as a unit, showed that, for instance, with a limited yield ($\bar{Q}_{well} = 0.4$) and the capacity of the underground reservoir, i.e. the reserve of water in the well ($\bar{W}_u = 0.1$), a tank with a capacity $\bar{W} = 0.56$ (or 33.6 m^3) should be installed, and with total regulation of the water supply rates, $\bar{W}$ increases up to 0.93 (or 56 m^3).

In the majority of cases, windmills are installed on already existent water sources, the water reserves in which are usually small. Therefore, in many cases one should orient oneselve primarily to the yield. With an increase in the relative yield, the capacity of an underground cistern required for total provision of water consumption is decreased, i.e. a smaller water reserve in the source is required. /259

In practice, with the exception of cases in new aquicultural construction, it is not possible to affect the value of W_u and it is possible to achieve an increase in the effectiveness of the installation by varying the value W, whose relationship with values of W_u and P' shows that an increase in W_u effects a decrease in W only up to a definite limit. A further increase in W_u has almost no effect on the value of W (Fig. 142).

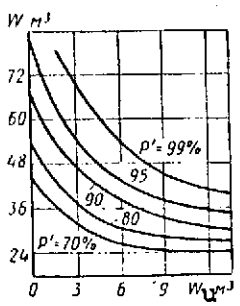


Fig. 142. Effect of water source fullness W_u and norm of mechanized water supply provision P' on the volume of a reservoir with B_d = = 12 m^3 and q_d = = 19.2 m^3 (according to the data of O. B. Khellenov).

The use of the coefficient of conformity of the graphs of loading and power output $\eta_{z.g} = (A_0 - A_z)/A_0$ in wind power-producing calculations as applicable to a wind pumping installation is not very promising, since the ouput, under set wind and hydrogeological conditions, is determined by factors which do not go into the expression for determining this coefficient (limited yield and coefficient of stoppages not connected with the wind rate and others).

The coefficient of utilization of established power $K_e = A_f/(N_eT)$, is the relationship of the actual power output by the machine to the output which is might give working for the entire period T at full established power N_e, simplifies calculations and allows additional graphic dependencies to be used for roughly determining power output [36].

For pumping installations, the use of coefficient K_e in calculations does not

give sufficient accuracy, since the actual output of power does not determine the quantity of water raised, as the result of which the working characteristics of the machine $Q(v)$ depend on its type, and K_e does not consider this characteristic. In practice we are talking only about the portion used of the energy of the airstream /260 passing through the surfaces swept by the windmill. Therefore, as when determining optimal parameters of a machine, calculations of wind installations with the required precision should be conducted with the method of sequential comparison of variations, which gives the indices of usable power output or the quantity of water which a machine equipped with a reservoir, can raise with a fixed rate for the chronological course of wind velocities.

Balanced calculations of wind machines with reservoirs are conducted according to previously recorded repetitions of velocities, from which the average periodic velocity $\overline{v}_{per}$ is usually revealed. Consequently, the norm of provisionability will be equal to 50%, i.e in five periods out of 10 the actual output may be lower than that calculated.

In hydrological calculations of reservoirs for hydro stations, the goal of establishing the optimal capacity of a reservoir to increase the output of a hydroelectric station was set up. Accumulating wind energy is possible only after the motor has used the kinetic energy of the stream. Therefore, no kind of reservoir (storage device) will enhance an increase in the possible total output, but only eases conditions of using the water (power), increasing provisionability of the consumer. Finally, relief and macrogeographic conditions frequently have an essential effect on the rate and intensity of the wind. In connection with this, to achieve more reliable power production indices, it is necessary to conduct additional investigations of the wind rate in each localized region.

Balancing calculations of wind installations are usually conducted in relative units; water consumption is taken as an independent variable, and its provision is the final goal for utilization of the wind's energy. Then, graphs of relative productivity $\overline{Q}(v,H)$, where $\overline{Q} = Q/B_d$ can be constructed for each machine (Fig. 143). By analogy, we introduce the concepts of relative values of calculated productivity $\overline{Q}_c = Q_c/B_d$, yield $\overline{q} = q/B_d$ and reservoir capacity $\overline{W} = W/B_d$. The calculations are conducted using differential-integral curves.

According to the algorithm and program developed (Fig. 144), calculations of reservoir volumes for wind machines were conducted using electronic computer equipment according to data of a 10-year series (period) of observations of wind velocity for the Kazakhstan and Povolzh'ye regions. Data from meteorological stations according to four wind velocity measurements per day were used, since sample comparison showed that the major aquicultural parameters according to these data and according the hourly wind

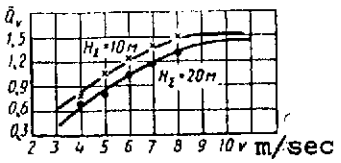


Fig. 143. Change in relative productivity of wind machine $\bar{Q}_v = Q_v/B_d$.

velocity measurements were usually in satisfactory agreement between each other.

On Fig. 145, a-c, showing the change in reservoir volumes in dependency on the degree of provision of mechanized water supply, /261 the type of machine, wind rates, etc., it is evident that one of the most important factors is the sequence of wind velocities, and not the value for average periodic velocities $\bar{v}_{per}$, although naturally with all other conditions being equal, the required capacity of a storage vessel is reduced in areas with higher values for velocity v_{per} (Table 29). The large dispersion σ of reservoir volumes attracts attention to itself.

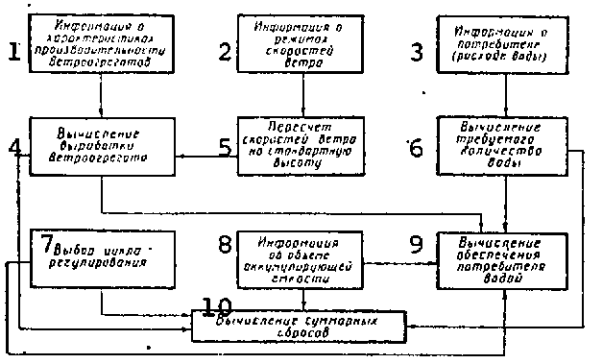


Fig. 144. Consolidated block diagram of program for calculating storage containers for pumping wind machines.

Key: 1. Information on characteristics of productivity of wind machines; 2. Information on wind velocity rates; 3. Information on consumption (water expenditure); 4. Compute output of wind machine; 5. Recalculate wind velocities according to standard height; 6. Compute required quantity of water; 7. Select cycle of regulation; 8. Information on volume fo storage containers; 9. Compute provision of water consumption; 10. Compute total surpluses.

A transfer from a semiannual to an annual cycle of adjustment will, as a rule, lead to an increase in the required volumes of reservoirs, although this increase frequently proves to be small. The value of the minimum working velocity of the wind v_0 has an essential effect on the volume of the reservoir W: its reduction from 4 to 3 m/sec decreases the required reservoir volume by 40-50%. On the other hand, with the characteristics of the machine and the consumer being unchanged, an increase in the norm of provision of a mechanized water supply above 95% will lead to a sharp increase in capacity W. This in turn reduces the value of water surplus S, which is caused by limited reservoir volume.

For instance, in regions with an average annual wind velocity $\bar{v}_a$ = 5-5.1 m/sec when using the VB-3T machine, with B_d = 6 m3,

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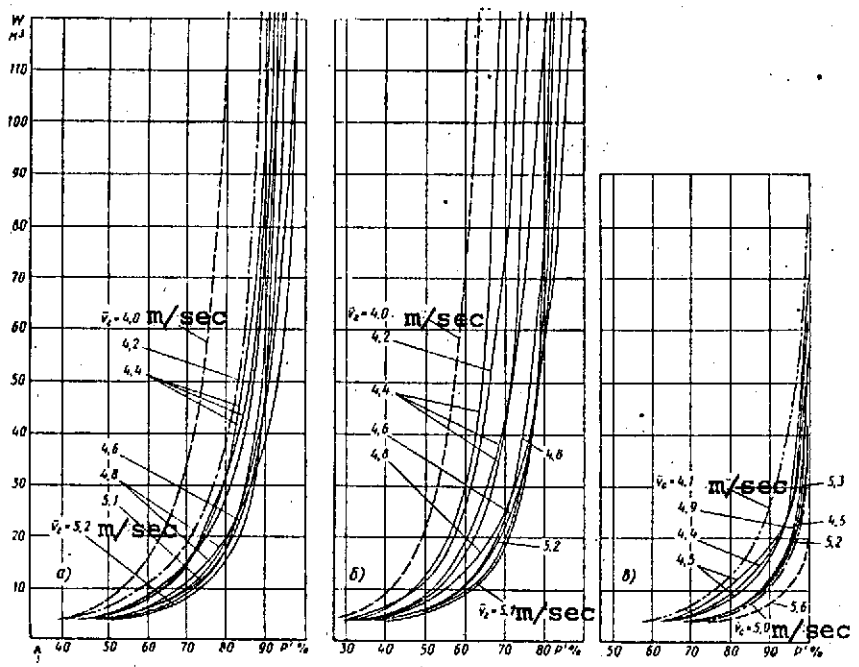

Fig. 145. Required capacity of pumping wind installation reservoirs: a. VTL-3 machine (T = 1 year; B_a = 3800 m^3); b. "Berkut" machine (T = 1 year; B_a = 3800 m^3); c. "Vikhr'" machine (T = 6 months; B_d = 8.8 m^3).

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TABLE 29. DEPENDENCY OF RESERVOIR VOLUMES IN m^3 ON DEGREE OF WATER SUPPLY PROVISION (ASTRAKHAN', 1950-1959)

Machine	Parameter	Period of adjustment: April-September v_{av} = 4.4 m/sec					October-March v_{av} = 4.8 m/sec					Year v_{an} = 4.6 m/sec				
		Degree of provision P' in %														
		70	75	80	85	90	70	75	80	85	90	70	75	80	85	90
TVM-3	m_W	8	10	15	24	40	5	6	8	10	16	6	8	11	16	26
	σ	2,4	5,1	9,7	21,2	36,3	1,4	1,9	2,3	3,9	8,5	1,6	4,4	4,8	8,6	19,0
2VPL-4	m_W	11	15	22	36	49	7	9	12	16	22	9	12	17	24	39
	σ	5,4	8,7	15,8	31,8	32,4	2,3	3,6	5,2	5,8	10,0	3,4	5,6	8,3	14,0	29,8
VB-3T	m_W	8	10	15	21	33	6	7	10	13	18	7	9	12	17	24
	σ	2,7	6,4	5,6	13,1	20,6	1,7	2,7	3,8	5,2	5,4	2,1	3,2	6,0	8,0	13,3
BNP-4M	m_W	6	9	12	18	29	5	6	8	10	14	6	8	10	14	20
	σ	4,2	3,5	5,9	10,5	14,4	0	1,8	2,5	3,9	5,0	1,7	1,7	4,0	6,3	11,0

it is necessary to equip the watering point with a tank having a volume of 8-25 m^3 with P' = 95-99% respectively over the course of the entire year. With B_d = 12 m^3, tanks having a volume of 21-46 m3 respectively are required (Fig. 146). This raises the question of the advisability in a number of cases of installing a machine with a high calculated productivity in order to decrease the reservoir volume with an increased provision of water supply. From the graphs it is evident that for P' > 93-95%, the required reservoir volumes increase sharply.

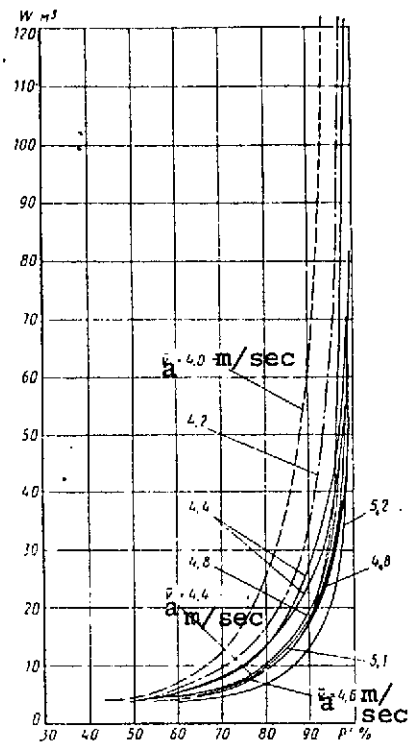

Fig. 146. Calculated values for volumes of reservoirs for a VB-3T pumping machine during its year-round utilization at a watering point with B_a = 3800 m^3.

We note the following conformity: /264 with a decrease in the values of average periodic wind velocity $\bar{v}_{per}$, the effect of water consumption on the volume of the tank, as a rule, increases. With $\bar{v}_{per}$ = 6-7 m/sec for a watering point with B_d = 6 m^3, a tank with a volume of up to 10 m^3 is needed, and with B_d = 12 m^3, a tank with a volume of 16-20 m^3 is needed. At a watering point with a "Buran" machine and with a tank having a volume of 18-20 m^3, with a daily water consumption in winter of 8.8 m^3 and in summer of 12 m^3 in regions with $\bar{v}_{per}$ = 6-7 m/sec at year-round pastures, mechanized water supply is provided at 95-96%. With seasonal utilization of sources in areas with $\bar{v}_d$ = 6.3 m/sec, a tank with a volume of 5-10 m3 allows P' = 80-90% to be obtained; desiring to obtain P' = 95%, it is necessary to increase the volume within limits of 17-22 m^3.

The average periodic wind velocity does not in full measure determine the required volume of the reservoir. This is connected with the prevailing effect of the overall tendency of variability of the wind in the actual zone, the character of the daily and seasonal course of its velocities, and sequencing of working and nonworking periods. With identical velocity $\bar{v}_{per}$, changes in reservoir volumes with P' = const are highly significant; they decrease as the intensity of the wind rate increases.

Fig. 147 shows the dependency $W(P', \bar{v}_{per}, V, T)$, and also the effect of the type of machine. In regions where the wind velocity

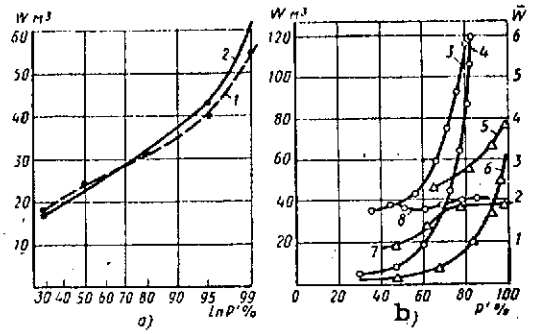

Fig. 147.-Change in required reservoir capacities depending on various factors: a. VP-3M machine (B_d = 5 m^3; H = 22 m; $\bar{q}$ = 4); b. adjustment of water consumption -- annual ($\bar{v}_a$ = 5 m/sec; $\bar{W}$ = (W VBL-3)/ /W; Δ -- B_a = 2000 m^3; o -- B_a = 4000 m^3); 1. calculated values; 2. experimental values; 3, 5. $\bar{W}$ for VB-3T machine; 4, 6. W for BVL-3 machine; 7, 8. $\bar{W}$ for VTL-3 machine.

of $\bar{v}_{per} \leq 4$ m/sec, an attempt to attain high provisionability of water supply through only the wind machine will lead to a significant increase in reservoir volume and adjusted expenditures. With velocities of $\bar{v}_a$ = 4.5-5.5 m/sec, which are characteristic for the major zones of herd livestock, high levels of water supply provision using the wind machine (90-98%) can be assumed. /265

Investigations conducted by the author together with engineer M. A. Bondarev for the conditions of Astrakhan, show that with a monthly cycle of water supply adjustment, a clear relationship between the values of $\bar{v}_{mon}$ and W is not present, especially in the area having a value of P' > 85%. With T = 1 month, a sharp increase in reservoir volume in this zone was not observed, and the curves rise monotonally, which makes selection of an optimal variation difficult.

Like with annual ones, with smaller adjustment cycles, an increase in water consumption by twice requires an increase in reservoir volume by 2.8-3.5 times. This relationship depends little on the value of P'. In general, mechanization using stationary well installations which are used 1 to 2 months per year are frequently not economical: it is more profitable to use light portable machines with internal combustion engines.

The effect of the characteristics of a wind machine on the required value of the storage device is extremely essential: thus, with T = 1 month for machines with identical power (VB-3T, VTL-3 and VBL-3) under identical conditions the values for W are relative as L : (1.7-2.2) : (3.5-4.1). These relationships are primarily affected by the values of v_0 and the shape of curve Q(v) of the machine. With T = 1 year and with one flock of sheep (B_a = 1900 m^3) for the VB-3T and "Vikhr'" machine, the following reservoirs are required: W = 5-7 m^3 (with P' = 80%), W = 8-10 m^3 (with P' = 90%) and W = 12-16 m^3 (with P' = 95%). With two flocks, the volumes are 15-16; 32-34 and 60-67 m^3 respectively. For a watering point with a VBL-3 machine and two flocks, a reservoir with a capacity of W = 76 m^3 (with P' = 80%) is required, and with one flock W = 16-17 m^3 is required. /266

Transfer from T = 1 month to T = 2 months will lead to a significant increase in the required reservoir capacity. Thus, with P' = 85%, H = 22 m and B_d = 6.5 m^3, this increase amounts to 42-49% (depending on the productivity of the machine. A further increase in the period of adjustment will have a smaller effect on the volume of the reservoir. If the reservoir volume is determined in accordance with an adjustment period for the season with the least intensive wind, the computed volume will secure reliable operation of the point for the course of the entire year [39, 58].

The yield of the source and the height of water feed, as well as the characteristics of the machine have an essential effect on the required volume of the reservoir (Tables 30, 31).

TABLE 30. REQUIRED RESERVOIR VOLUME IN m^3 FOR WATERING POINT WITH VP-3M MACHINE ($\bar{v}_{per}$ = 2.5 m/sec)

(a) Высота подъема воды в м	(b) Норма обеспечения P' в %					
	75			85		
	(c) Суточный дебит водоисточника в $м^3$					
	6,5	13	19,5	6,5	13	19,5
5	27	19	18	33	23	21
10	53	31	29	64	37	36
15	78	39	37	90	48	47
20	90	47	42	104	59	56

Key: (a) Height of water lift in m
(b) Norm of provision P' in %
(c) Daily yield of water source in m^3

TABLE 31. REQUIRED RESERVOIR VOLUME IN m^3 (B_d = 13 m^3; H_Σ = 14 m; $\bar{v}_a$ = 2.9 m/sec)

Wind machine	Water source yield in m^3								
	$q = 1{,}35B_d$			$q = 2B_d$			$q = 4B_d$		
	Norm of provision P' in %								
	75	85	95	75	85	95	75	85	95
VP-3M	170	192	238	44	52	67	17	19	24
VB-3	126	146	183	37	41	48	16	18	22
"Buran"	172	198	248	39	47	61	16	18	23

From data in the tables it is evident that at low-yield sources in areas where $\bar{v}_a \leq 3$ m/sec, due to the large required volume of reservoirs, wind machines are usually not very effective, to the point that optimal norms of mechanized water supply provision are low (no higher than 80%). In order to decrease the volume of reservoirs, it is necessary to try to install machines primarily on sources with a high yield. In areas with $\bar{v}_a$ = 3-4 m/sec, with P' = 75-95%, for economic considerations, volumes of W turn out to be acceptable only for a yield $q \geq 2B$. In regions with $v_{per} > 4$ m/sec, the requirement for yield is reduced: the yield q_{min} must be no less than (1-.4-1.5)B. With smaller values of yield, the only effective method for improving economic indices of the installation is increasing the water reserve in the well. As the investigations of O. B. Khellenov showed, if $q/B > 3$, a further increase in yield has only a slight effect on reservoir volume (Fig. 148).

The consumer always strives to obtain guaranteed water supply with a norm of provision of P' = 100%. This is theoretically impossible, particularly for a wind pumping machine. It is possible, however, to attain 98-99% provisionability, but expenses for raising water with this are increased. The method of variational calculations provides the possibility in each actual case for establishing an economically feasible and technically possible limit of the norm of provision P'. In the majority of cases, the limit does not exceed 95%.

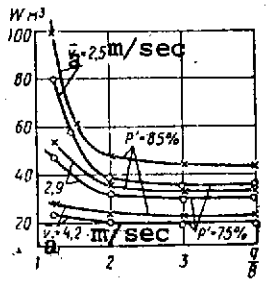

Fig. 148. Effect of water source yield on the require capacity of a reservoir for the VB-3T machine with B_d = 5 m³, H_Σ = 22 m.

With a limited yield, the coefficient of utilization of the possible output of the installation K_δ is decreased in areas with high values for average velocities $\bar{v}_{per}$.

If the yield increases by twice, coefficient K_δ increases by 30-60% (for various machines).

Comparison of calculated and empirical data according to the "Bashlak" watering point (Turkmenian SSR) showed the high resemblance (the difference in the value of W is now greater than 5.6%). An analytical expression of the dependency W(P') for P' = 65-95% has the appearance of an exponential function [39]

$$W = a_1 + be^{a_2 P'}. \qquad (273)$$

The values of coefficients for a given machine depend on repetition of wind velocities, characteristics of the source and the norm of water consumption. An important preserve in reducing

the required volume of a storage tank is utilization of wind machines with as wide a possible range of working wind velocities. With this, the value of v_0 does not, as optimization calculations showed, have to exceed 3.5 m/sec.

Looking at a pumping wind installation as a complex, consisting /268 of the machine, storage reservoir and water source, and considering that the indices of this complex are effected chiefly by the value and graph of water consumption, and intensity and singularities of the wind rate in the area, special attention should be paid to the selection of each of the elements of the complex with fixed initial conditions. This signifies the necessity for comparing a number of possible variations and adopting the one of them which yields the minimum adjusted expenditures.

If the machine does not have a storage device then, for instance, with a double watering of livestock the graph of water consumption for P = 90° in the conditions of the Kyzyl-Kum desert, might be filled out through operating the machine without adjusting devices (reservoirs) by an average of 40-45%. Consequently, the remaining 60-55% of the water requirement, in accordance with the loading graph, must be provided through reserves which are always in readiness [32].

In the majority of cases this is unacceptable because of technical-economic considerations, and in pasture conditions the advisability of using a wind machine in general falls under doubt, since along with additional expenditures for doubling the installation, there must be a servicing crew which is constantly located near the installation to bring in fuel and so forth. With this, the coefficient of utilization of the established power of the non-wind engine proves to be extremely low. The use of high capacity electrochemical batteries and water supply installations /269 is difficult. These reasons, experience from operation and completion of calculations show that water supply with a set norm of provisionability is most effective in the presence of a reserve tank.

With a set pressure head, the productivity of the machine depends only on wind velocity. However, limitations caused by an insufficient yield in the source and disagreement between the supply rate and consumption of water with a limited reservoir capacity can be superposed on productivity. Consequently, the total output of the machine is

$$Q_{\Sigma} = f(\bar{v}_{per}, q, H, W, B). \qquad (274)$$

It has been established that $P' < 80\%$ should not be taken due to the periodic disruptions in the schedule of water

consumption and the necessity for frequently drawing in manual labor; with P' > 97%, the volumes of reservoirs prove the be sufficiently high so that expenditures on raising the water increase essentially. Besides this, the construction and operation of reservoirs in pasture conditions is accompanied by known difficulties. Consequently, it is necessary to find the optimal reservoir guaranteeing provision of the required water consumption and yielding the minimum adjusted costs with that established water consumption. With the impossibility of using the entire output of the machine, calculations for this reservoir are conducted according to the value of B.

If the capacity of the reservoir is insufficient, losses from a deficient water supply will cause spoilage in the produce at the consumer's or a reduction in its quantity. Naturally, objective evaluation of this loss with a sufficient degree of reliability is difficult as the result of the compilation of the effects of many factors which are not functionally connected with the wind rates and work of the machine. In turn, exposing the effect of the factors indicated is also not easy. With an inadmissible decrease in provision of power or water, however, it is necessary for periods of extended calms to raise water manually or in rare cases (for instance, with high pressure heads H_{Σ}) to use an internal combustion engine. With this, losses arise due to the additional expenditures on fuel, servicing and so forth.

For selecting the best variation of a wind installation from the point of view of optimal reservoir volume and, consequently, minimal expenses, it is convenient to set up a dependency of calculated expenses on reservoir volume for the given conditions. We assume that with volume W_1 the mechanized supply of water by the machine is provided at 80%, and with W_2, it increases up to 90%. This means that it is necessary to raise 20% of the water manually or by another means in the first case, and 10% in the second case. At small objects, as a rule, manual labor or animals will be used. The total output of the machine over a period T will be $Q_{\Sigma T} = BT$. The optimal variation of the installation can be determined according to a system (Table 32).

Calculations for a case in which the deficit in water supply /270 is made up using a non-wind motor can also be performed according toa similar system.

As a result of the fact that water consumption has been taken as an independent parameter fixed for a given consumer, construction of graphs is more convenient in the form $\bar{E}(W,P')$. From Fig. 150 it is evident that minimal expenses in the case examined are obtained in the interval P' = 86-91%, which corresponds to W_{opt} = 55-69 m^3. However, it is also necessary to keep in mind those cases in which the dominating factor is not expenditures $\bar{E}$, but, for instance, the impossibility of using manual labor, the

TABLE 32. DETERMINATION OF OPTIMAL STORAGE CAPACITY

Determining value	Variation with provision P'_1 (reservoir capacity W_1)	Variation with provision P'_2 (reservoir capacity W_2)
Deficit in water supply D in m^3 over period T, which must be made up with another source of power	$D_1 = BT\left(\frac{100 - P'_1}{P'_1}\right)$	$D_2 = BT\left(\frac{100 - P'_2}{P'_2}\right)$
Output of machine advantageously used in m^3	$Q_{n_1} = BT\frac{P'_1}{100}$	$Q_{n_2} = BT\frac{P'_2}{100}$
Capital investments on building the installation in rubles	$K_{W_1} = K_a + K_n + \beta_1 W_1$	$K_{W_2} = K_a + K_n + \beta_2 W_2$
Expenditures $\bar{E}$ for raising water with the wind machine in rubles/m^3	$\bar{E}_{W_1} = \frac{U_{T_1}100}{BTP'_1} + \varphi_n \bar{K}_{W_1}$	$\bar{E}_{W_2} = \frac{U_{T_1}100}{BTP'_2} + \varphi_n \bar{K}_{W_2}$
Expenditures $\bar{E}$ on nonmechanized water supply to cover the deficit in rubles/m^3	$\bar{E} = \bar{C}_{P_1}$	$\bar{E} = \bar{C}_{P_2}$
Average expenditures in rubles/m^3	$\bar{E}_1 = \frac{\bar{E}_{W_1} Q_{n_1} + \bar{E}_n D_1}{BT}$	$\bar{E}_2 = \frac{\bar{E}_{W_1} Q_{n_1} + \bar{E}_n D_2}{BT}$

Note. Here K_a, K_n -- capital investments for the machine's acquisition and repair respectively, in rubles; β is the coefficient reflecting the specific cost of a reservoir in a definite capacity interval and depending on its design (Fig. 149).

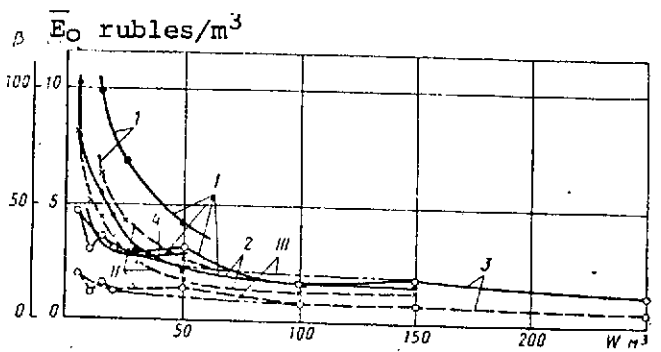

Fig. 149. Specific capital investments β and operational expenditures $\bar{E}_o$ for water pressure towers and reservoirs (according to data of typical plans): 1. brick tower, h_t = 10 m; 2. brick reservoir; 3. reinforced concrete reservoir; 4. ferrous cement reservoir; I. β per 1 m^3 of reservoir capacity; II. $\bar{E}$; III. expected changes in β and $\bar{E}_o$ for reinforced concrete reservoirs.

absence of fuel and so forth. Considering these considerations for conditions, it is sometimes necessary to use not the optimal, but higher values for reservoir volume.

Thus, it has been established that reservoir volumes are determined, based on information on hand concerning performance of the calendar sequence of wind velocities over the past period. With a change in only one parameter (B, Q, P' and others), it is necessary to conduct a new cycle of calculations. The

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difficulty in using analytical and probability and statistical methods consists of the fact that the process of chronological changeability of the wind over a extended time interval (season, year), is random and apparently cannot be considered stationary without extremely significant admissions. The curves $v(t)$ do not possess the property of erosion.

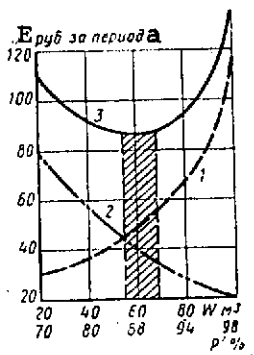

Fig. 150. Effect of reservoir capacity on expenditures:
1. expenditures on raising water with a wind machine;
2. expenditures on raising water manually or with a reserve engine;
3. expenses on the entire volume of water supply over period T.

Key: a. E rubles over the period.

The task of finding the volume of a reservoir will be simplified if sufficiently accurate models for determining groups of curves $v(t)$, which are standardized according to the proper size beforehand, are worked out (see Chapter 1). Then for determining the value of W, it is necessary to use expressions (269) and (270), which allow the deficits in power, its losses (water surpluses) and other parameters characterizing storage to be found for each moment in time.

Solution of this problem, preferably in relative units, allows recommendations for selection of values W and E of the storage capacities to be nomographed in their overall form as applicable to actual regional an economic conditions.

When calculating electric storage batteries, their coefficients of efficiency must be considered, and they do not usually exceed 80%. This means that the output of power by the machine must be 20% higher than its consumption by the consumer. We also note that when calculating reservoirs it is desirable to consider water losses due to vaporization (4-5% of the total consumption).

The consumption of the power of the duplicating motor ($N_{d.m}$) causes no smaller difficulties. it is not expedient, in the majority of cases, to take it as equal to the established power of the wind machine N_e, since the coefficient of loading of the duplicating motor and its average operation coefficient of efficiency turn out to be low. When supplying power to several current receivers, the maximum total (peak) power of all consumers which must be provided with power according to the set schedule are usually considered, and the duplicating (reserve) electric station is also selected according to it. Only in cases where the wind machine provides power to a single consumer, whose power N_c is equal to N_e of the machine and the use of storage devices is impossible, it is

$N_{d.m} - N_e$

Chapter 12. Calculation of Technical and Economic Indices of Wind Installations and their Regional Use /272

§ 1. Technical and Economic Indices of Wind Installations

The singularities in calculating technical and economic indices of wind installations are connected with the probable character of change of the power source and the power developed, the effect of storage facility capacity, and for pumping machines, also the yield of the water source. Therefore, the methodology for calculation is based on determination of the total power-producing effect from use of the installation and expenditures which are necessary for this.

The energy-producing effect is defined as the total output of energy A over the calculated period or the quantity of water raised, which is figured on the basis of many years' data on repetition of wind velocities. If the entire output cannot be used, total water or power consumption is introduced into the calculation.

The energy of the wind is not profitably utilized everywhere and, in order to answer whether or not the use of a wind installation is expedient in the actual case, it is necessary to compare its economic indices with the indices of other power-producing installations which could be used in the given conditions. Comparing calculated expenses of a number of possible variations for solving the given power-production problem, i.e. examining the interchangeable variations providing indentical satisfaction of the consumer with power or water, the most advantageous solution is established. In comparison with a non-wind installation, it is also necessary to maintain conditions of approximate equality of power, since a number of specific indices depend on this.

We will examine the methodology for calculation with an example of pumping machines. The effectiveness of the installation depends on characteristics of the motor and the period of its utilization, the wind rate, water consumption, depth and yield of the well, set provisionability of water supply, and also the system for technical servicing.

Knowing the dependency $Q(v)$ of the installation according to manufacturer's data and knowing repetition of wind velocities according to a distribution curve, we determine the total theoretical output $Q_{\Sigma T}$ which is the maximum one according to expression (269). It is possible also to use equation (91), then transferring from energy to quantity of water raised (see Chapter 4). The real possible or actual output $Q_{\Sigma a}$ is usually smaller than $Q_{\Sigma T}$, part of the time with active wind velocities the motor stands still due to

repairs and technical servicing, during the night hours, as a result of limited yield, storage capacities and other reasons. The reduction in output is considered with statistical coefficient k_o and k_r (see Chapter 4). If $q > Q_{max}$ and the volume of the reservoir is sufficiently great, then $k_r = 1$.

It is usually difficult to select a machine in which /273
$Q_{\Sigma a} = BT$, and therefore the calculations should be conducted according to water consumption, equating output to it. Calculation according to actual output $Q_{\Sigma a}$ is conducted when the water is also used for irrigating a section near the well and for other purposes.

The area of preferred use for one installation or another is determined by constructing combination graphs giving the dependency $\bar{E}(\bar{v}, Q_\Sigma)$ (Fig. 151). In some cases, similar constructions for cost of water and capital investments may be required. Designating $\bar{E}$, K and C with indices w, i and e as adjusted expenditures, capital investments and cost of water respectively for wind, internal combustion and electrical installations, we obtain

$$\bar{E}_{w(i,e)} = \rho_n \frac{K_{w(i,e)}}{Q_\Sigma(B_\Sigma)} + \frac{U_{w(i,e)}}{Q_\Sigma(B_\Sigma)} \text{ rub/m}^3, \qquad (275)$$

where U is the annual (seasonal) operation expenses (outlays); ρ_n is the normative coefficient of effectiveness (for power-producing installations, $\rho_n = 8$ years is usually taken.

An installation in which expenditures are $\bar{E}_{min}$ is the most advantageous for set conditions, only if special conditions do not impose any sort of limitation on its utilization.

The cost of lifting water is

$$C_a = \frac{U_d}{Q_{\Sigma a}} \text{ or } C_d = \frac{U_d}{Q_{\Sigma d}} \text{ rub/m}^3 \qquad (276)$$

and the period of return of additional capital investments in the construction of the wind installation by comparison with other means for mechanizing water supply is

$$T_{add} = \frac{K_w - K_{i(e)}}{S} = \frac{K_w - K_{i(e)}}{(\bar{E}_{i(e)} - \bar{E}_w)Q_\Sigma} \text{ years}, \qquad (277)$$

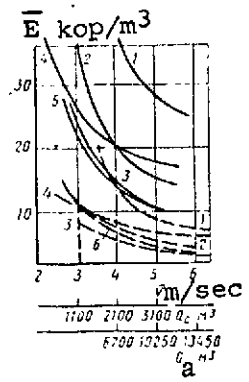

Fig. 151. Specific calculated expenditures on mechanization of water lifting for pastures with seasonal (solid lines) and year-round (broken lines) utilization of the installation: 1. electric installation with a distance from the line of 2 km; 2. electric installation with a distance from the line of 1 km; 3. electric installation with a distance from the line of 0.5 km; 4. internal combustion pumping station; 5. VP-3M water lift; 6. "Buran" machine.

where S is the annual (seasonal) savings in rubles.

At first glance, a dependency between the average periodic velocity of the wind $\bar{v}_{per}$ and expenditures for lifting water with an internal combustion or electric installation does not have to exist, but since the wind motor gives a high output at high velocities $\bar{v}_{per}$, a decrease in expenditures $\bar{E}$ and cost C takes place. The output of all installtions is taken as identical, and therefore for them, $\bar{E} = f(\bar{v}_{per})$. /274

Expenses in time on servicing a wind motor installed on a farm are taken as two manhours per day; on servicing an internal combustion, it is taken as 50% of its working time; and on servicing an electric installation, 20 min for each engagement and disengagement (together with inspection and preparation for starting). The wages of a category IV mechanic who services the installation amount to 34 kop/h. The field raise for a mechanic of pasture installations is considered. With group servicing of installations by a riding mechanic, the wages, expenses on fuel and depreciation deductions from the cost of the automobile are applied proportionally to the number of installations. According to the experience of a number of organizations, 6-8 h per month are expended on servicing one machine. The norm of amortized deductions from the cost of equipment is taken according to Table 33.

When using an electric pump as a water source which is removed from a high voltage line by up to 1 km, it is necessary to consider the capital investments only for a low voltage line; at greater distance, the cost of a high voltage line and a transformer substation also go into the calculation. Expenditures for ongoing repair of the transformer substation and low voltage line are equal to 3%; that of a high voltage one is 5% of their balanced cost.

In typical installations for raising water in pastures, a belt water lift with a ZID-4.5 motor or an electric pump powered by a ZhES-2 electric station is usually used.

TABLE 33. NORMS OF ANNUAL DEDUCTIONS FOR RECONDITIONING AND REPAIR

Equipment	Deduction in percent of balanced cost		Equipment	Deduction in percent of balanced cost	
	For reconditioning	For ongoing and major repairs		For reconditioning	For ongoing and major repairs
Well	1.7	2.0	Internal combustion engine	18.8	13.75
Reinforced concrete tank	2.0	1.5	Belt water lift	13.2	8.56
Watering trough	2.0	1.5	GAZ-63 motor vehicle	18.0	3.46*
Wind machine	7.2	5.0			

* Per each 10,000 km of travel.

Regardless of high capital investments for construction of a wind installation by comparison with an internal combustion one, the wind installation is more economic due to the low operational expenditures. By comparison with manual pumping, expenses for its construction are recouped over several months; by comparison with internal combustion installations, expenses are recouped over 1-2 years. /275

According to the graphs (in Figs. 151 and 152) it is easy to establish in which conditions the use of one type of installation or another is profitable. For instance, with year-round utilization, the "Buran" machine is more profitable than an internal combustion installation in regions where $v_a \geq 3.1$ m/sec; an installation powered from a line which passes at a distance $l \leq 0.5$ km from the well, with $v_a < 6.7$ m/sec is more economic than a wind machine. With $l = 1$ km, the electric station is more profitable only where $v_a < 3.1$ m/sec, and with $l \geq 2$ km, the wind machine is always more economic than water raising installations of other types.

If water consumption is limited, capital investments on non-wind installations, adjusted expenditures and cost of raising 1 m^3 of water do not depend on the average wind velocity. Capital investments and expenditures on wind pumping installations are

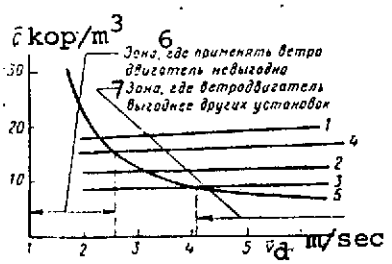

Fig. 152. Cost of raising water at a watering point with a daily water consumption of 10 m3 (seasonal utilization of installation): 1-4. same as in Fig. 151; 5. wind motor with belt water lift; 6. zone where use of a wind motor is not profitable; 7. zone where a wind motor is more profitable than other installation

reduced in areas with high velocities, since with B = const, motors of lower power are required (Fig. 152).

Under actual conditions, it is necessary to consider the number and qualifications of the persons necessary for servicing installations which are guaranteed by the factory to perform a number of hours of the motor's work without major overhaul and so forth. Thus, for pasture wind installations, the constant presence of servicing personnel is not required, which is an important circumstance. The service life of wind machines is significantly longer than the service life of internal combustion engines, and therefore specific expenditures of metal per unit of output over the entire service life in them are lower.

Table 34 presents economic indices of an internal combustion installation and a "Vikhr'" machine with the grouped system of servicing.

In areas with wind velocity $\bar{v}_a \geq 3.5$ m/sec, expenditures on raising water in pastures using a wind installation are lower than with other means, and, besides this, its servicing is simpler. This leads to the important conclusion that a wise and effective system of water supply in pastures, and also for some farms and other objects in areas not serviced with electricity requires the widespread usage of wind installations. This conclusion is supported by data from state testing of machines in northern Kazakhstan (Table 35) and with results of calculations performed by O. B. Khellenov, for conditions of Turkmenia (Table 36). /276

It is more profitable to use installations on year-round pastures with full utilization of the entire output of the machine. The cost of 1 m3 of water with H_Σ = 22 m does not exceed 12 kopecks ($\bar{v}_a$ = 4.5 m/sec). With a daily water consumption B_d = 10 m3, it increases up to 14 kop/m3, and with B_d = 5 m3, up to 22 kop/m3 (depending on wind conditions). With half-year utilization of the mechanization means, expenditures increase by 25-30% by comparison with utilization of the machine over the course of the entire year. If H_Σ = 14-15 m (conditions most characteristic for pastures), expenses are reduced by 6-10% by comparison with the expenses with H_Σ = 25 m.

TABLE 34. COST AND ADJUSTED EXPENDITURES FOR RAISING WATER (H_Σ = 20 m)

$\bar{v}_a$ in m/sec	"Vikhr'" machine				(a) Насос с ЖЭС-2	
	$Q_{\Sigma_a} = Q_{max}$		$Q_{\Sigma_a} = 2200$ м³		$Q_{\Sigma_a} = 2200$ м³	
	C in kop/m³	E in kop/m³	C in kop/m³	E in kop/m³	C in kop/m³	E in kop/m³
4	4.8	7.9	9.1	15	18.6	23
5	3.7	6.1	9.1	15	18.6	23
6	3.2	5.3	9.1	15	18.6	23

Key: (a) pump with ZhES-2

TABLE 35. ECONOMIC INDICES OF WIND MACHINES IN AREAS WITH WIND VELOCITIES $\bar{v}_a$ = 4.5 m/sec ($Q_\Sigma = Q_{w\ max}$)

Indices	VTL-3	VBL-3	VEN-4	VB-3T
Total pressure head in m	22	23	26	17
Expenses of labor in man-h/m³	0,0036	0,0045	—	0,0096
Specific capital investments in rub/m³	0,49	0,6	0,61	0,24
Cost of raising water in rub/m³	0,11	0,14	0,097	0,029

Captial expenses on mechanization using wind machines by comparison with a nonmechanized method (even for small water consumptions) are recouped in the majority of cases over the course of 2 years. In seasonal pastures in areas with a wind velocity $\bar{v}_d \leq 3$ m/sec with the same water consumptions, the period of repayment exceeds the normative. Additional capital investments by comparison with belt water lifts are recouped in less than 4 years, regardless of the duration of the period of utilization with $\bar{v}_{per} \geq 2.5$ m/sec. This is explained by the decrease in the require reservoir capacity with an increase in wind intensity and with a reduction in expenses for transport means for a riding mechanic with grouped servicing. /277

A comparison of water supply variations conducted by the USSR Giprovodkhoz [All-Union State Planning, Surveying and Scientific Research Institute of Water Management Construction] showed that the most economic was a watering point with a wind motor (Table 37). Expenditures on watering sheep were reduced by 4-5.4 times by comparison with averages across the country. /278

With the servicing of 12 machines by one mechanic, indices for their utilization in areas with wind velocity $\bar{v}_{per}$ = 5 m/sec are given in Table 38.

TABLE 36. TECHNICAL AND ECONOMIC INDICES OF RAISING WATER IN THE PASTURES OF TURKMENIA (B = 5 m^3/day)

Indices	Installations			VLM-100 water lift with 2SD-v motor	Raising water with a camel pull
	VP-3M	VB-3T	2VPI-4		
	v_{per} in m/sec				
	2,9	4,2	4,2		
Capital investments in watering point, including the well, in rub	4031	3550	3950	3586	3128
Specific capital expenditures, in kop/m^3	9,5 / 38,0	8,3 / 33,2	9,6 / 38,3	12,2 / 49,0	6,2 / 24,6
Cost of water in kop/m^3	21,2 / 44,6	19,8 / 39,6	21,4 / 45,0	37,2 / 60,5	52,2 / 76,6
Adjusted expenditure in kop/m^3	30,7 / 82,5	28,1 / 72,8	31,0 / 83,3	49,4 / 109,5	58,9 / 101,2
Expenses of labor in man-h/m^3	0,14	0,14	0,14	0,8	1,0
Productivity of labor in m^3/man-h	7,2	7,2	7,2	1,25	1,0
Reduction in labor expenses by comparison with a nonmechanized method of raising water in %	86	86	86	20	—
Reduction of direct expenses by comparison with nonmechanized method in %	58	62	59	29	—
Period of recovery in years of additional capital investments by comparison with the VLM-100 water lift	1,3 / 3,7	0,12 / 0,3	1,3 / 3,7	—	—

Notes: 1. The numerator shows numbers for installations used round the year, the dominator shows seasonal ones; 2. One driving mechanic-driver services 10 wind installations or four belt water lifts.

TABLE 37. SOME RELATIVE ECONOMIC INDICES OF WATER SUPPLY IN PASTURES WITH A WATERING RADIUS OF 5 km (H_Σ = 50 m; B_d = 12 m^3)

Indices	Независимое водоснабжение (a)			Групповое (e) водоснабжение при транспортировании воды	
	VP-3M	Ленточный водоподъемник с приводом (b)			
		(c) от двигателя ЗИД-4,5	(d) от передвижной ЖЭС	(f) по трубопроводам	(g) автоцистернами
Capital investments in %	100	108	188	432	83
Operation expenditures in %	100	188	576	110	800
Cost of water in %	100	177	453	261	1230

Key: (a) Independent water supply
(b) Belt water lift with drive
(c) From ZID-4.5 engine
(d) From portable ZhES
(e) Grouped water supply with transportation of water
(f) Along pipelines
(g) With tankers

TABLE 38. INDICES OF PUMPING WIND INSTALLATIONS

Object	T in days	β in m^3/days	Machine	K in thousand rubles	v_a	c in rum/m^3
Watering point	365	15 15	"Berkut" "Vikhr'"	3,54 3,43	0,388 0,365	0.07 0,06
Field station	210	2 2	"Berkut" "Vikhr'"	2.38 2,16	0,333 0,315	0,53 0,5

Energy from low power, high-speed electric wind machines used for charging storage batteries and illumination in regions with a wind velocity v_a = 4.5-5.5 m/sec, costs 4-9 kop/kW·h, depending on the capacity of the battery, the power of the consumer and the schedule of loading.

Attention should be paid to the fact that the values for actual Q_a and calculated Q_c productivity do not always coincide (Fig. 153). This can be explained by the structure of changeability of velocities within limits of the interval between periods of measuring velocities (1.3 or 6 h). productivity Q_a is actually higher than Q_c, especially with low values of $v$per. This is caused by a disagreement between the methods of measuring and treating velocities used and the character of utilizing the wind to obtain energy.

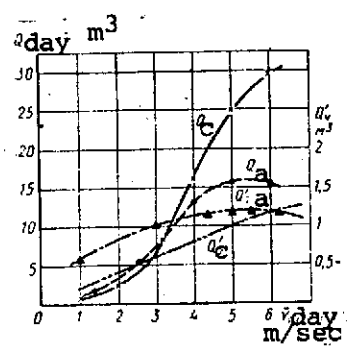

Fig. 153. Change in actual and calculated values of average daily output Q_a and Q_c and hourly productivity Q_h and Q_c of a VP-3M wind machine [39].

Usually, curves of repetition of $v(t)$, according to which calculations are made, are compiled according to data of hourly or periodic observations. With this, a velocity is used /279 which is averaged over a selected time interval and does not consider the dynamics of change in velocity within that interval. With this approach, it is necessary to consider that if v_t = = a, the power of the machine proportional to v^3, will be $N = ka^3$, and output over time t_a of action of this velocity is

$$Q_{t_i} = ka^3 t_a \text{ м}^3. \qquad (278)$$

Summarizing outputs with consideration for the duration of energy-producing calms and periods when the machine works with regulated power,

gives the calculated value for the maximum possible output $Q_{\Sigma c}$ over the entire period.

In reality, however, the measured velocity $\bar{v}_t$ is compiled of velocities $v_i \leq \bar{v}_t$ and $v_i > \bar{v}t$, and for each value of $\bar{v}_t$, a dispersion depending on Δt can be found. For instance, with $\bar{v}h$ = 5 m/sec, the range (v_{min}-v_{max}) = 3-7 m/sec. The interval of averaging for so-called "momentary" velocities of the wind is determined by the mechanical characteristics of the machine, primarily the moment of inertia J_{wm}.

The power of the motor is determined by the "momentary" velocity, and therefore the actual output is

$$Q_{\Sigma a} = k\left(\frac{v_{i1}^3 t_1 + v_{i2}^3 t_2 + \cdots + v_{in}^3 t_n}{n}\right) м^3. \tag{279}$$

As a rule, it must be higher than $Q_{\Sigma c}$, since

$$\left(\frac{a+b+c+\cdots+n}{t_n}\right)^3 \leqslant \frac{a^3+b^3+c^3+\cdots+n^3}{t_n}. \tag{280}$$

The deviation of velocity v from the average decreases with an increase in the latter. In zones with an intensive rate of high velocity repetition, when the machine works with regulated power, lower deviations between productivity $Q_{\Sigma a}$ and $Q_{\Sigma c}$ can be expected. For the steppe and semidesert areas of Central Asia and Kazakhstan with $\bar{v}_a$ = 5 m/sec, productivity $Q_{\Sigma a}$ is greater than $Q_{\Sigma c}$ by 15-17%. /280

According to data of investigations performed in Turkmenia, the coefficients of correlation

$$K_c = \frac{\sum xy - n\bar{y}\bar{x}}{\sqrt{(\sum x^2 - n\bar{x}^2)(\sum y^2 - n\bar{y}^2)}}, \tag{281}$$

where x = $Q_{\Sigma a}$; y = $Q_{\Sigma c}$ are with various values of $\bar{v}_{day}$; x, y are the mathematical expectations; n is the number of observations, for calculated and actual productivity $Q_{\Sigma c}$ and $Q_{\Sigma a}$ are relatively high (up to 0.95), and for calculated and actual productivity of work t_c and t_a, as should be expected, they are low (about 0.55).

§ 2. Regional Use of Wind Machines

The type and characteristics of a machine must be selected with consideration for the wind, hydrogeological and economic

conditions of the area. If, for instance, a low-speed wind motor, whose calculations for strength were performed for relatively low velocities is installed in an area with high wind intensity, its longevity is expected to be deficient. These motors are usually regulated by turning the windmill out of the wind and a significant portion of the time with wind velocities above 17 m/sec, they will be standing. In areas with high wind velocities and high gustiness of the flow, low-speed motors operate unstably. The use of a high-speed motor without a relieving system in zones with low values for velocity v_{per} will lead to a reduction in the coefficient of utilization of working time and in the effectiveness of the installation.

Selection of the water lift has no less meaning. It must correspond to the characteristics of the source: depth, yield, volume of water collection part and water makeup. For instance, many pit wells have water with high sand and salt content, and therefore normal piston pumps rapidly break down and measures must be taken (chroming the cylinder walls, installing cuffs of special rubber, filters) so as to increase their service life. For these wells it is better to use belt lifts, pneumatic, screw and inertia pumps. On the contrary, for fill-in wells, belt water lifts cannot be used due to their mixing of fresh water with saline; here pneumatic pumps or other water lifts with laminar water collection in the surface layer should be installed. Since during the pumping of water the thickness of the fresh layer is decreased, the pump must be fastened so that it cannot take in saline water. It must be fed on the surface only as the fresh layer is renewed, and the rest of the time the pump will work at idle or it should be stopped. Under these conditions, centrifugal or screw pumps cannot be used: they rapidly break down. /281

For raising water from bore and small-pipe wells, it is more advantageous to use machines with a screw pump, inertia water lift or with a submerged centrifugal pump. If the source has a good filter and the water does not contain sand, installations with piston pumps are also effective.

Electric pumping wind machines, created during recent years, can be used with total pressure heads of up to 100 m, and installations with piston and screw pumps can be used with static heads of up to 40 m, those with inertia water lifts -- for depths of up to 20 m, and pneumatic and belt lifts are used for wells with a depth of 30-40 m.

In pastures, the total head with which the machine works is usually 15-20% higher than the height from the head of a well or the top of a reservoir to the dynamic level of water. When supplying water to farms, the pressure necessary for pumping water into a tower is also considered. Usually $H_{\Sigma} = H_{st} + (15\text{-}20)$ m. It is recommended that a machine with some reserve in pressure be used, since during the summer period the level of water in the well

frequently drops. It is necessary that the maximum productivity of the machine is less than the yield of the well.

The type of wind machine is determined primarily by the wind rate, characteristics of the consumers, and the average daily consumption of water or power.

The coastal areas of the north, Caspian coast and the northern part of Sakhalin are characterized by high intensity of wind rate ($\bar{v}_a > 6$ m/sec), and hurricane winds (over 30 m/sec) accompanied by snow blizzards and storms, are frequently observed here. In this area it is better to use machines with high-speed motors, whose strength is calculated with $v_{max} = 40$ m/sec.

Many areas of the European part of RSFSR, Khazakhstan, Central Asia (with the exception of the southern zones) and regions southeast of Lake Baykal have a velocity $\bar{v}_a = 4$-6 m/sec. With a velocity $\bar{v}_a$ up to 5 m/sec, $v_{max} < 25$ m/sec and a low wind gustiness (Ukraine, Belorussia, a large part of Turkmenia and Uzbekistan, the center of the European part), motors with $Z_n = 2$-5 with any systems for orientation and limitation of power are well recommended. Higher speed machines with relieving systems can also be used.

In the Kazakh SSR, Azerbaydahzn SSR and in some zones of the RSFSR (Black earth, Kizlyarsk patures, Povolzh'ye and others) where an average annual wind velocity is higher, the wind is gusty and hurricanes are accompanied with sand and dust storms, machines with high speed, with a wind rose system of orientation and with windmills having fully rotating blades should be used. With air temperatures below -10°C, it is difficult to operate installations with belt water lifts, since the well head requires proper heating. /282

Eastern Siberia and the Far East, some areas of the European part, a large part of Tadzhikistan, the southern regions of the Kazakh and Uzbek SSR and the southeastern part of Turkmenia have, as a rule, low values for average annual and average seasonal wind velocities, and here widespread usage of wind machines is not recommended. However, there are regions where as the result of relief conditions and other factors, the wind rate in certain seasons is better, and low-speed motors can be effectively used.

When selecting a machine, it is necessary to consider the seasons of its utilization and climactic conditions of the region, the system of servicing, transport communications in the region and the place for installing the motor.

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